STEAM BOILERS

Their CONSTRUCTION
CARE and OPERATION

With Questions and Answers

SWINGLE

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STEAM BOILERS

THEIR CONSTRUCTION, CARE AND OPERATION

With Questions and Answers

A RELIABLE UP TO DATE TREATISE, CONTAINING FULL DESCRIPTIONS, WITH ILLUSTRATIONS OF THE VARIOUS TYPES OF BOILERS, INCLUDING ALL THE LEADING TYPES OF WATER TUBE BOILERS IN USE TODAY. A LARGE SECTION IS DEVOTED TO BOILER CONSTRUCTION, GIVING RULES AND FORMULÆ FOR ASCERTAINING THE STRENGTH OF RIVETED SEAMS; ALSO METHODS OF BRACING. INSTRUCTIONS FOR SETTING BOILERS, EITHER BY THE ORDINARY METHOD, OR BY SUSPENSION. RULES FOR FINDING SAFE WORKING PRESSURE; ADJUSTMENT OF SAFETY VALVES, ASCERTAINING HEATING SURFACE, MAKING EVAPORATION TESTS, ETC. COMBUSTION CLEARLY EXPLAINED. ALL LEADING TYPES OF MECHANICAL STOKERS DESCRIBED AND ILLUSTRATED.

By CALVIN F. SWINGLE, M. E.

AUTHOR OF "ENCYCLOPEDIA OF ENGINEERING," "EXAMINATION QUESTIONS AND ANSWERS FOR MARINE AND STATIONARY ENGINEERS," "MODERN LOCOMOTIVE ENGINEERING," ETC.

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INTRODUCTION

Inestimable benefits have been conferred upon the human race through the medium of steam in its varied applications to the needs of men, but too often the steam generator itself is neglected, through ignorance or carelessness, and allowed to deteriorate until it becomes an actual source of danger instead of a useful agent.

There is a constant and growing demand in the manufacturing world for more power from steam, and as the boiler is the source of this power, too much attention cannot be paid to its construction, care and management.

It cannot be denied that a great responsibility rests upon any man who has charge of a boiler or battery of boilers, and it should be the chief aim of a man in such a position to thoroughly acquaint himself with his boilers, their construction, care, and the safest and most economical method of operating them. It has been the aim of the author, in compiling the following pages, to place before the student some facts regarding steam boilers which he believes will be helpful to engineers.

Boiler construction has been dwelt upon at considerable length, especially the strength and efficiency of riveted joints.

The staying of flat surfaces, including all the various kinds of stays, receives particular attention also, for the reason that in the opinion of the author a man taking charge of a boiler should first of all be compe-

tent to ascertain, by means of certain standard rules and formulas, what is the safe working pressure of his boiler, also the proper area of the safety valve in proportion to grate area, and many other details of bracing, riveting, etc.

Boiler settings and appurtenances, feed pumps and injectors, boiler operation, washing out, etc., all receive due attention.

Combustion, the nature of steam and correct rules for making evaporation tests are all treated upon at length, also the subject of firing, both by hand and by the use of mechanical stokers.

The highest authorities have been consulted in the make-up of the book, and this fact, coupled with the author's practical experience of thirty five years as an engineer, it is believed, will make the book of great value to its readers.

C. F. S.

Steam Boilers

CHAPTER I

THE BOILER

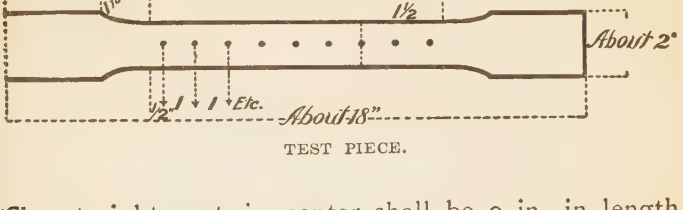
Importance of correct knowledge of the construction and strength of steam boilers—Tensile strength of steel boiler plates—Dr. Thurston's specifications—Specifications of U. S. board of inspectors of steam vessels—Punched and drilled plates—Rivets and rivet iron and steel—Efficiency of the joints—Proportions of double riveted butt joints—Lloyd's rules for thickness of plate and diameter of rivets—Correct design of triple riveted butt joints—Calculations for efficiency of different forms of joints—Discussion of various ways in which failure may occur in different styles of joints—Necessity for higher efficiencies in riveted joints—Quadruple and quintuple butt joints—Staying flat surfaces—Different methods of staying a boiler—Correctly designed stays—Stay bolts for fire box boilers—The Belpaire boiler—Vanderbilt Locomotive with Morison fire box—Gusset stays—Through stay rods—Calculating strength of stayed surfaces—Area of segments—Proper spacing of stays—Strength of unstayed surfaces—Dished heads—Welded seams.

As it is of the highest importance, not only to the engineer in charge of the plant, but also to his assistants, and in fact to all persons whose business compels them to be in the vicinity of the boiler-room, that there should be absolutely no doubt as to the safe construction of the boilers and their ability to withstand the pressures under which they are operated, the

author has compiled the following valuable data regarding the construction and strength of boilers. The deductions and reports of tests and experiments made by such eminent authorities as Dr. Thurston, Prof. Wm. Kent, Dr. Peabody, D. K. Clark, Hutton and many other experts have been consulted, and the author has also added the results of his own observations, collected during an experience of thirty-five years as a practical engineer.

When steel was first introduced as a material for boiler plate, it was customary to demand a high tensile strength, 70,000 to 74,000 lbs. per sq. in., but experience and practice demonstrated in course of time that it was much safer to use a material of lower tensile strength. It was found that with steel boiler plate of high tenacity there was great liability of its cracking, and also of certain changes occurring in its physical properties, brought about by the variations in temperature to which it was exposed. Consequently present-day specifications for steel boiler plate call for tensile strengths running from 55,000 to 66,000 lbs., usually 60,000 lbs. per sq. in. Dr. Thurston gives what he calls "good specifications" for boiler steel as follows: "Sheets to be of uniform thickness, smooth finish, and sheared closely to size ordered. Tensile strength to be 60,000 lbs. per sq. in. for fire box sheets and 55,000 lbs. per sq. in. for shell sheets. Working test: a piece from each sheet to be heated to a dark cherry red, plunged into water at 60° and bent double, cold, under the hammer. Such piece to show no flaw after bending. The U. S. Board of Supervising Inspectors of Steam Vessels prescribes, in Section 3 of General Rules and Regulations, the following method for ascertaining the tensile strength of steel plate for

boilers: "There shall be taken from each sheet to be used in shell or other parts of boiler which are subject to tensile strain, a test piece prepared in form according to the following diagram:



TEST PIECE.

The straight part in center shall be 9 in. in length and 1 in. in width, marked with light prick punch marks at distances 1 in. apart, as shown, spaced so as to give 8 in. in length. The sample must show, when tested, an elongation of at least 25 per cent in a length of 2 in. for thickness up to ¼ in. inclusive; in a length of 4 in., for over ¼ in. to $\frac{7}{16}$ in. inclusive; in a length of 6 in., for all plates over $\frac{7}{16}$ in. and under 1¾ in. in thickness. The samples shall also be capable of being bent to a curve of which the inner radius is not greater than 1½ times the thickness of the plates, after having been heated uniformly to a low cherry red and quenched in water of 82° F."

Punched and Drilled Plates. Much has been written on this subject, and it is still open for discussion. If the material is a good, soft steel, punched sheets are apparently as strong and in some instances stronger than drilled, especially is this the case with regard to the shearing resistance of the rivets, which is greater with punched than with drilled holes.

Concerning rivets and rivet iron and steel Dr.

Thurston has this to say in his "Manual of Steam Boilers": "Rivet iron should have a tenacity in the bar approaching 60,000 lbs. per sq. in., and should be as ductile as the very best boiler plate when cold. A good 5/8-in. iron rivet can be doubled up and hammered together cold without exhibiting a trace of fracture." The shearing resistance of iron rivets is about 85 per cent and that of steel rivets about 77 per cent of the tenacity of the original bar, as shown by experiments made by Greig and Eyth. The researches made by Wöhler demonstrated that the shearing strength of iron was about four-fifths of the tensile strength.

The tables that follow have been compiled from the highest authorities and show the results of a long and exhaustive series of tests and experiments made in order to ascertain the proportions of riveted joints that will give the highest efficiencies.

The following Table 11 gives the diameters of rivets for various thicknesses of plates and is calculated according to a rule given by Unwin.

TABLE 11

TABLE OF DIAMETERS OF RIVETS*

Thickness of Plate	Diameter of Rivet	Thickness of Plate	Diameter of Rivet
1/4 inch	1/2 inch	9/16 inch	7/8 inch
5/16 "	9/16 "	5/8 "	15/16 "
3/8 "	11/16 "	3/4 "	1 1/16 "
7/16 "	3/4 "	7/8 "	1 1/8 "
1/2 "	13/16 "	1 "	1 1/4 "

The efficiency of the joint is the percentage of the strength of the solid plate that is retained in the joint,

*Machine design—W. C. Unwin.

and it depends upon the kind of joint and method of construction.

If the thickness of the plate is more than ½ in., the joint should always be of the double butt type.

The diameters of rivets, rivet holes, pitch and efficiency of joint, as given in the following Table 12, which was published in the "Locomotive" several years ago, were adopted at the time by some of the best establishments in the United States.*

TABLE 12

PROPORTIONS AND EFFICIENCIES OF RIVETED JOINTS

	Inch	Inch	Inch	Inch	Inch
Thickness of plate	1/4	5/16	3/8	7/16	1/2
Diameter of rivet	5/8	11/16	3/4	13/16	7/8
Diameter of rivet-hole	11/16	3/4	13/16	7/8	15/16
Pitch for single riveting	2	2 1/16	2 1/8	2 3/16	2 1/4
Pitch for double riveting	3	3 1/8	3 1/4	3 3/8	3 1/2
Efficiency—single-riveted joint	.66	.64	.62	.60	.58
Efficiency—double-riveted joint	.77	.76	.75	.74	.73

Concerning the proportions of double-riveted butt joints, Prof. Kent says: "Practically it may be said that we get a double-riveted butt joint of maximum strength by making the diameter of the rivet about 1.8 times the thickness of the plate, and making the pitch 4.1 times the diameter of the hole."

Table 13 as given below is condensed from the report of a test of double-riveted lap and butt joints.† In

*Thurston's "Manual of Steam Boilers."

†Proc. Inst. M. E., Oct., 1888.

this test the tensile strength of the plates was 56,000 to 58,000 lbs. per sq. in., and the shearing resistance of the rivets (steel) was about 50,000 lbs. per sq. in.

TABLE 13

DIAMETER AND PITCH OF RIVETS—DOUBLE-RIVETED JOINT

Kind of Joint	Thickness of Plate	Diameter of Rivet	Ratio of Pitch to Diameter
Lap	$\frac{3}{8}$ inch	0.8 inches	3.6 inches
Butt	$\frac{3}{8}$ "	0.7 "	3.9 "
Butt	$\frac{3}{4}$ "	1.1 "	4.0 "
Butt	1 "	1.3 "	3.9 "

Lloyd's rules, condensed, are as follows:

LLOYD'S RULES—THICKNESS OF PLATE AND DIAMETER OF RIVETS

Thickness of Plate	Diameter of Rivets	Thickness of Plate	Diameter of Rivets
$\frac{5}{8}$ inch	$\frac{5}{8}$ inch	$\frac{3}{4}$ "	$\frac{7}{8}$ inch
$\frac{7}{16}$ "	$\frac{5}{8}$ "	$\frac{13}{16}$ "	$\frac{7}{8}$ "
$\frac{1}{2}$ "	$\frac{3}{4}$ "	$\frac{7}{8}$ "	1 "
$\frac{9}{16}$ "	$\frac{3}{4}$ "	$\frac{15}{16}$ "	1 "
$\frac{5}{8}$ "	$\frac{3}{4}$ "	1 "	1 "
$\frac{11}{16}$ "	$\frac{7}{8}$ "		

The following Table 14 is condensed from one calculated by Prof. Kent,* in which he assumes the shearing strength of the rivets to be four-fifths of the tensile strength of the plate per square inch, and the excess strength of the perforated plate to be 10 per cent.

* Kent's "Mechanical Engineer's Pocket-Book," page 362.

TABLE 14

Thickness of Plate	Diameter of Hole	Pitch		Efficiency	
		Single Riveting	Double Riveting	Single Riveting	Double Riveting
Inches	Inches	Inches	Inches	Per Cent	Per Cent
3/8	7/8	2.04	3.20	57 1	72.7
7/16	1	2.30	3.61	56.6	72.3
1/2	1	2.14	3.28	53 3	70.0
1/2	1 1/8	2.57	4.01	56.2	72.0
9/16	1	2.01	3.03	50.4	67.0
9/16	1 1/8	2.41	3.69	53.3	69.5
9/16	1 1/4	2.83	4.42	55.9	71.5
5/8	1	1.91	2.82	47.7	64.6
5/8	1 1/8	2.28	3.43	50.7	67.3
5/8	1 1/4	2.67	4.10	53.3	69.5

Another table of joint efficiencies as given by Dr. Thurston* is as follows, slightly condensed from the original calculation:

TABLE 15

Single riveting

Plate thickness.	5/16''	3/8''	7/16''	1/2''	5/8''	3/4''	7/8''	1''
Efficiency	.55	.55	.53	.52	.48	.47	.45	.43

Double riveting

Plate thickness.	3/8''	7/16''	1/2''	3/4''	7/8''	1''
Efficiency.....	.73	.72	.71	.66	.64	.63

The author has been at considerable pains to compile Tables 16, 17 and 18, giving proportions and efficiencies of single lap, double lap and butt, and triple-riveted butt joints. The highest authorities have been consulted in the computation of these tables and great care exercised in the calculations.

*Thurston's "Manual of Steam Boilers," page 119.

TABLE 16

PROPORTIONS OF SINGLE-RIVETED LAP JOINTS

Thickness of Plate Inches	Diameter of Rivet Inches	Pitch of Rivet Inches	Efficiency Per Cent
5/16	9/16	1.13	50.5
"	5/8	1.33	53.3
"	11/16	1.55	55.7
3/8	3/4	1.60	53.3
"	7/8	2.04	57.1
7/16	7/8	1.87	53.2
"	1	2.30	56.6
1/2	1	2.14	53.3
"	1 1/8	2.57	56.2
9/16	1	2.01	50.4
"	1 1/8	2.41	53.3
"	1 1/4	2.83	55.9
5/8	1 1/8	2.28	50.7
"	1 1/4	2.67	53.3

It will be noticed that in single-riveted lap joints the highest efficiencies are attained when the diameter of the rivet hole is about 2 1/3 times the thickness of the plate, and the pitch of the rivet 2 5/8 times the diameter of the hole.

With the double-riveted joint it appears, according to Table 17, that in order to obtain the highest efficiency the joint should be designed so that the diameter of the rivet hole will be from 1 3/4 to 2 times the thickness of plate, and the pitch should be from 3 1/3 to 3 1/2 times the diameter of the hole. Concerning the thickness of plates Dr. Thurston has this to say:* "Very thin plates cannot be well caulked, and thick plates cannot be safely riveted. The limits are about 1/4 of an inch for the lower limit, and 3/4 of an inch for the higher limit." The riveting machine, however, overcomes the difficulty with very thick plates.

* Thurston's "Manual of Steam Boilers," page 120.

TABLE 17

PROPORTIONS OF DOUBLE-RIVETED LAP AND BUTT JOINTS

Thickness of Plate	Diameter of Rivet	Pitch of Rivet	Efficiency
5/16 inch	9/16 inch	1.71 inches	67.1 per cent
5/16 "	5/8 "	2.05 "	69.5 "
3/8 "	3/4 "	2.46 "	69.5 "
3/8 "	7/8 "	3.20 "	72.7 "
7/16 "	3/4 "	2.21 "	66.2 "
7/16 "	7/8 "	2.86 "	69.4 "
7/16 "	1 "	3 61 "	72.3 "
1/2 "	1 "	3.28 "	70.0 "
1/2 "	1 1/8 "	4.01 "	72.0 "
9/16 "	1 "	3.03 "	67.0 "
9/16 "	1 1/8 "	3.69 "	69.5 "
9/16 "	1 1/4 "	4.42 "	71.5 "
5/8 "	1 1/8 "	3.43 "	67.3 "
5/8 "	1 1/4 "	4.10 "	69.5 "
3/4 "	1 "	2.50 "	72.0 "
7/8 "	1 1/8 "	3.94 "	74.2 "
1 "	1 1/4 "	4.10 "	76.1 "

The triple-riveted butt joint with two welts, one inside and one outside, has two rows of rivets in double shear and one outer row in single shear on each side of the butt, the pitch of rivets in the outer rows being twice the pitch of the inner rows. One of the welts is wide enough for the three rows of rivets each side of the butt, while the other welt takes in only the two close pitch rows.

When properly designed, this form of joint has a high efficiency, and is to be relied upon. Table 18 gives proportions and efficiencies, and it will be noted that the highest degree of efficiency is shown when the diameter of rivet hole is from 1¼ to 1½ times the thickness of plate, and the pitch of the rivets is from 3½ to 4 times the diameter of the hole. This, of

course, refers to the pitch of the close rows of rivets, and not the two outer rows.

TABLE 18

PROPORTIONS OF TRIPLE RIVETED BUTT JOINTS WITH INSIDE AND OUTSIDE WELT

Thickness of Plate Inches	Diameter of Rivet Inches	Pitch of Rivet Inches	Pitch of Outer Rows Inches	Efficiency Per Cent
3/8	13/16	3.25	6.5	84
7/16	13/16	3.25	6.5	85
1/2	13/16	3.25	6.5	83
9/16	7/8	3.50	7.0	84
5/8	1	3.50	7.0	86
3/4	1 1/16	3.50	7.0	85
7/8	1 1/8	3.75	7.5	86
1	1 1/4	3.87	7.7	84

Some simple rules are given in Chapter III for finding the percentage of efficiency, or in other words the ratio of the strength of the riveted joint to the strength of the solid plate. In those calculations the tensile strength of the rivets was assumed to be 38,000 to 40,000 lbs. per sq. in. The highest efficiency is attained in a riveted joint when the tensile strength of the rods from which the rivets are cut approaches that of the plates, and when the proportions of the joint are such that the tensile strength of the plates, the shearing strength of the rivets, and the crushing resistance of the rivets and plate, for a given section or unit strip, are as nearly equal as it is possible to secure them.

A few examples of calculations for efficiency will be given, taking the three forms of riveted joints in most common use. The following notation will be used throughout:

T.S. = Tensile strength of plate per square inch.
T = Thickness of plate.
C = Crushing resistance of plate and rivets.
A = Sectional area of rivets.
S = Shearing strength of rivets.
D = Diameter of hole (also diameter of rivets when driven).
P = Pitch of rivets.

In the calculations that follow T.S. will be assumed to be 60,000 lbs., S will be taken at 45,000 lbs., and the value of C may be assumed to be 90,000 to 95,000.

Fig. 97 shows a double-riveted lap joint. The style of riveting in this joint is what is known as chain riveting.

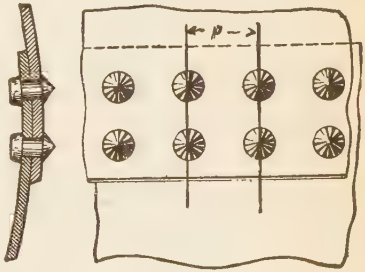


FIGURE 97.

In case the rivets are staggered, the same rules for calculating the efficiency will hold as with chain riveting, for the reason that with either style of riveting the unit strip of plate has a width equal to the pitch or distance p, Fig. 97.

The dimensions of the joint under consideration are as follows: P = 3¼ in., T = $\frac{7}{16}$ in., D = 1 in. (which is also diameter of driven rivet).

The strength of the unit strip of solid plate is P × T × T.S. = 85,312.

The strength of net section of plate after drilling is P − D × T × T.S. = 59,062.

The shearing resistance of two rivets is 2A × S = 70,686.

The crushing resistance of rivets and plate is $D \times 2 \times T \times C = 78{,}750$.

It thus appears that the weakest part of the joint is the net strip or section of plate, the strength of which is 59,062 and the efficiency = $59{,}062 \times 100 \div 85{,}312 = 69.2$ per cent.

A double-riveted butt joint is illustrated by Fig. 98, and the dimensions are as follows:

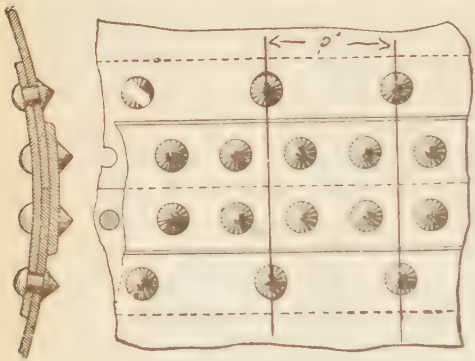


FIGURE 98.

P, inner row of rivets = 2¾ in.

P′, outer row of rivets = 5½ in.

T of plate and butt straps = $\frac{7}{16}$ in.

D of hole and driven rivet = 1 in.

Failure may occur in this joint in five distinct ways, which will be taken up in their order.

1. Tearing of the plate at the outer row of rivets. The net strength at this point is P − D × T × T.S., which, expressed in plain figures, results as follows: $5.5 - 1 \times .4375 \times 60{,}000 = 118{,}125$.

2. Shearing two rivets in double shear and one in single shear. Should this occur, the two rivets in the inner row would be sheared on both sides of the plate, thus being in double shear. Opposed to this strain there are four sections of rivets, two for each rivet. Then at the outer row of rivets in the unit strip there is the area of one rivet in single shear to be added.

The total resistance, therefore, is $5A \times S$ as follows: $.7854 \times 5 \times 45{,}000 = 176{,}715$.

3. The plate may tear at the inner row of rivets and shear one rivet in the outer row. The resistance in this case would be $P' - 2D \times T \times T.S. + A \times S$ as follows: $5.5 - 2 \times .4375 \times 60{,}000 + .7854 \times 45{,}000 = 127{,}218$.

4. Failure may occur by crushing in front of three rivets. Opposed to this is $3D \times T \times C$, or $1 \times 3 \times .4375 \times 95{,}000 = 124{,}687$.

5. Failure may occur by crushing in front of two rivets and shearing one. The resistance is represented by $2D \times T \times C + 1A \times S$; expressed in figures, $1 \times 2 \times .4375 \times 95{,}000 + .7854 \times 45{,}000 = 118{,}468$.

The strength of a solid strip of plate 5½ in. wide before drilling is $P' \times T \times T.S.$, or $5.5 \times .4375 \times 60{,}000 = 144{,}375$, and the efficiency of the joint is $118{,}125 \times 100 \div 144{,}375 = 81.1$ per cent.

A triple-riveted butt joint is shown in Fig. 99, the dimensions of which are as follows:

$T = \frac{7}{16}$ in.
$D = \frac{15}{16}$ in.
$A = .69$ in.
$P = 3\frac{3}{8}$ in.
$P' = 6\frac{3}{4}$ in.

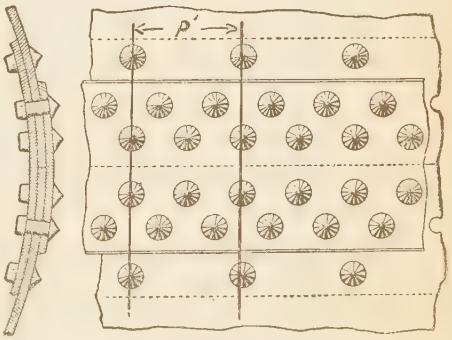


FIGURE 99.

Failure may occur in this joint in either one of five ways.

1. By tearing the plate at the outer row of rivets where the pitch is 6¾ in. The net strength of the unit

strip at this point is $P' - D \times T \times T.S.$, found as follows: $6.75 - .9375 \times .4375 \times 60{,}000 = 152{,}578$.

2. By shearing four rivets in double shear and one in single shear. In this instance, of the four rivets in double shear, each one presents two sections, and the one in single shear presents one, thus making a total of nine sections of rivets to be sheared, and the strength is $9A \times S$, or $.69 \times 9 \times 45{,}000 = 279{,}450$.

3. Rupture of the plate at the middle row of rivets and shearing one rivet. Opposed to this strain the strength is $P' - 2D \times T \times T.S. + 1A \times S$, equivalent to $6.75 - (.9375 \times 2) \times .4375 \times 60{,}000 + .69 \times 90{,}000 = 190{,}068$.

4. Crushing in front of four rivets and shearing one rivet. The resistance in this instance is $4D \times T \times C + 1A \times S$, or $.9375 \times 4 \times .4375 \times 90{,}000 + .69 \times 45{,}000 = 178{,}706$.

5. Failure may be caused by crushing in front of five rivets, four of which pass through both the inside and outside butt straps, while the fifth rivet passes through the inside strap only, and the resistance is $5D \times T \times C$, equivalent to $.9375 \times 5 \times 90{,}000 = 184{,}570$.

The strength of the unit strip of plate before drilling is $P' \times T \times T.S.$, or $6.75 \times .4375 \times 60{,}000 = 177{,}187$, and the efficiency is $152{,}578 \times 100 \div 177{,}187 = 86$ per cent.

With the constantly increasing demand for higher steam pressures, the necessity for higher efficiencies in the riveted joints of boilers becomes more apparent, and of late years quadruple and even quintuple-riveted butt joints have in many instances come into use. The quadruple butt joint when properly designed shows a

high efficiency, in some cases as high as 94.6 per cent. Fig. 100 illustrates a joint of this kind, and the dimensions are as follows:

T = ½ in.
D = $\frac{15}{16}$ in.
A = .69 in.
P, inner rows = 3¾ in.
P′, 1st outer row = 7½ in.
P″, 2d outer row = 15 in.

The two inner rows of rivets extend through the

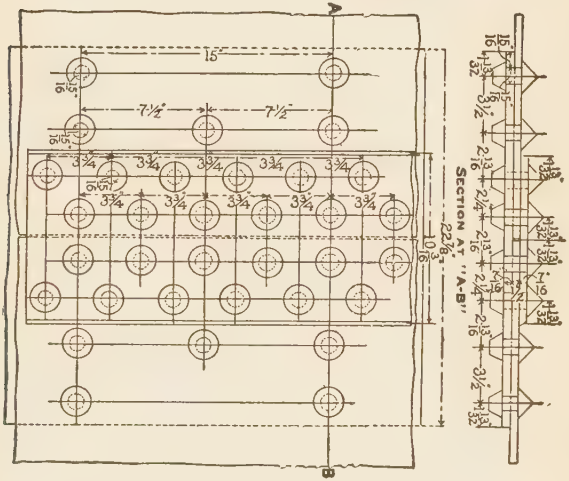


FIGURE 100.

main plate and both the inside and outside cover plates or butt straps.

The two outer rows reach through the main plate and inside cover plate only, the first outer row having twice the pitch of the inner rows, and the second outer row has twice the pitch of the first.

Taking a strip or section of plate 15 in. wide (pitch of outer row), there are four ways in which this joint may fail.

1. By tearing of the plate at the outer row of rivets. The resistance is $P' = D \times T \times T.S.$, or $15 - .9375 \times .5 \times 60{,}000 = 421{,}875$.

2. By shearing eight rivets in double shear and three in single shear. The strength in resistance is $19A \times S$, or $.69 \times 19 \times 45{,}000 = 589{,}950$.

3. By tearing at inner rows of rivets and shearing three rivets. The resistance is $P''' = 4D \times T \times T.S. + 3A \times S$, or $15 - (.9375 \times 4) \times .5 \times 60{,}000 + .69 \times 3 \times 45{,}000 = 430{,}650$.

4. By tearing at the first outer row of rivets, where the pitch is $7\frac{1}{2}$ in., and shearing one rivet. The resistance is $P'''' = 2D \times T \times T.S. + A \times S$, or $15 - (.9375 \times 2) \times .5 \times 60{,}000 + .69 \times 45{,}000 = 424{,}800$.

It appears that the weakest part of the joint is at the outer row of rivets, where the net strength is 421,875. The strength of the solid strip of plate 15 in. wide before drilling is $P''' = T \times T.S.$, or $15 \times .5 \times 60{,}000 = 450{,}000$, and the efficiency is $421{,}875 \div 450{,}000 = 93.7$ per cent.

Staying Flat Surfaces. The proper staying or bracing of all flat surfaces in steam boilers is a highly important problem, and while there are various methods of bracing resorted to, still, as Dr. Peabody says, "the staying of a flat surface consists essentially in holding it against pressure at a series of isolated points which are arranged in regular or symmetrical pattern." The cylindrical shell of a boiler does not need bracing for the very simple reason that the internal pressure tends to keep it cylindrical. On the contrary the internal pressure has a constant tendency to bulge out the flat

surface. Rule 2, Section 6, of the rules of the U. S. Supervising Inspectors provides as follows: "No braces or stays hereafter to be employed in the construction of boilers shall be allowed a greater strain than 6,000 lbs. per sq. in. of section."

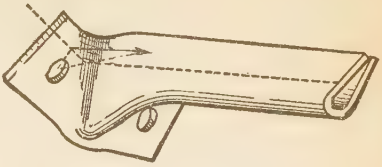


FIGURE 101.

The method to be employed in staying a boiler depends upon the type of boiler and the pressure to be carried. Formerly when comparatively low pressures were used (60 to 75 lbs. per sq. in.) the diagonal crow foot brace was considered amply sufficient for staying the flat heads of boilers of the cylindrical tubular type, both above and below the tubes, but in the present age, when much higher pressures are demanded, through stay rods are largely employed. These are soft steel or iron rods 1¼ to 2 in. in diameter, extending through from head to head, with a pull at right angles to the plate, thus having a great advantage over the diagonal stay in that the pull on the diagonal stay per square inch of section is more than 5 per cent in excess of what a through stay would have to resist under the same conditions of pressure, etc. The method of calculation for diagonal bracing is given in Chapter I and will not be discussed here.

FIGURE 102.

The weakest portion of the crow foot brace when in position is at the foot end, where it is connected to the head by two rivets. With a correctly designed brace

the pull on these rivets is direct and the tensile strength of the material needs to be considered only, but if the form of the brace is such as to bring the rivet holes above or below the center line of the brace, or if the rivets are pitched too far from the body of the brace, there will be a certain leverage exerted upon the rivets in addition to the direct pull. Fig. 101 shows a brace of incorrect design and Figs. 102 and 103 show braces designed along correct lines.

FIGURE 103.

Other methods of staying, besides the crow foot brace and through stays, consist of gusset stays, and for locomotives and other fire box boilers screwed stay bolts are employed to tie the fire box to the external shell. The holes for these stay bolts are punched or drilled before the fire box is put in place. After it is in and riveted along the lower edge to the foundation ring, or mud ring as it is sometimes called, a continuous thread is tapped in the holes in both the outside plate and the fire sheet by running a long tap through both plates. The steel stay bolt is then screwed through the plates and allowed to project enough at each end to permit of its being riveted cold. Stay bolts are liable to be broken by the unequal expansion of the fire box and outer shell, and a small hole should be drilled in the center of the bolt, from the outer end nearly through to the inner end. Then in case a bolt breaks, steam or water will blow out through the small hole, and the break will be discovered at once. The problem of properly staying the flat crown sheet of a horizontal fire box boiler, especially a locomotive

boiler, is a very difficult one and has taxed the inventive genius of some of the most eminent engineers.

Before the invention of the Belpaire boiler, with its outside or shell plate flat above the fire box, the only method of staying the crown sheet was by the use of cumbersome crown bars or double girders extending across the top of the crown sheet and supported at the ends by special castings that rest on the edges of the side sheets and on the flange of the crown sheet. At intervals of 4 or 5 in. crown bolts are placed, having the head inside the fire box and the nut bearing on a plate on top of the girder. There is also a thimble for each bolt to pass through, between the top of the crown sheet and the girder. These thimbles maintain the proper distance between the crown sheet and girder and allow the water to circulate freely.

The Belpaire fire box dispenses with girders and permits the use of through stays from the top of the flat outside plate through the crown sheet and secured at each end by nuts and copper washers.

For simplicity of construction and great strength the cylindrical form of fire box known as the Morison corrugated furnace has proved to be very successful. This form of fire box was in 1899 applied to a locomotive by Mr. Cornelius Vanderbilt, at the time assistant superintendent of motive power of the New York Central and Hudson River R. R. This furnace was rolled of ¾-in. steel, is 59 in. internal diameter and 11 ft. 2¼ in. in length. It was tested under an external pressure of 500 lbs. per sq. in. before being placed in the boiler. It is carried at the front end by a row of radial sling stays from the outside plate, and supported at the rear by the back head. Figs. 104 and 105 show respectively a sectional view and an end elevation of

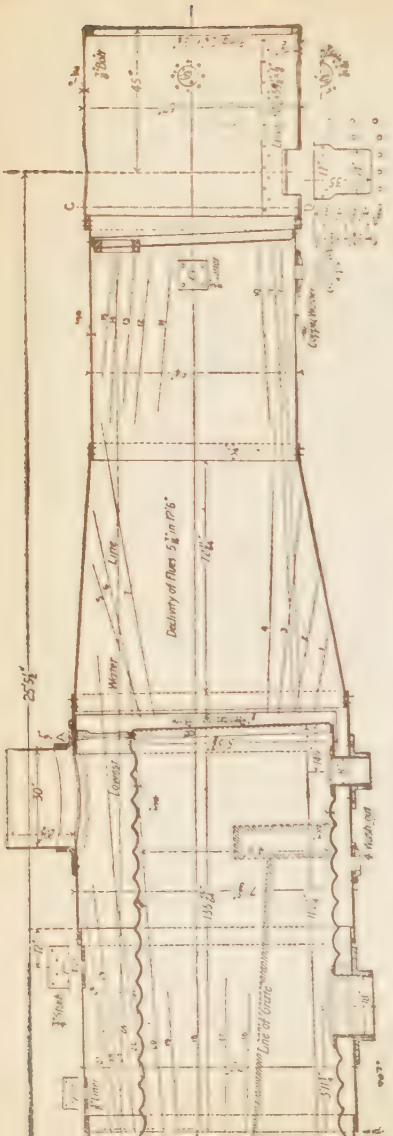


FIGURE 104.

this boiler. It will be seen at once that the question of stays for a fire box of this type becomes very simple. The boiler has proved to be so satisfactory that the company has since had five more of the same type constructed.

Gusset stays are used mainly in boilers of the Lancashire model and are triangular-shaped plates sheared to the proper form and having two angle irons riveted to the edges that come against the shell and the head. The angle irons are then riveted to the shell and the flat head. This form of brace is simple and solid, but its chief defect is, that it is very rigid

and does not allow for the unequal expansion of the internal furnace flues and the shell. Fig. 106 illustrates a gusset stay and the method of applying it.

Coming now to through stay rods, it is safe to say that whenever and wherever it is possible to apply them they should be used. In all cases they should

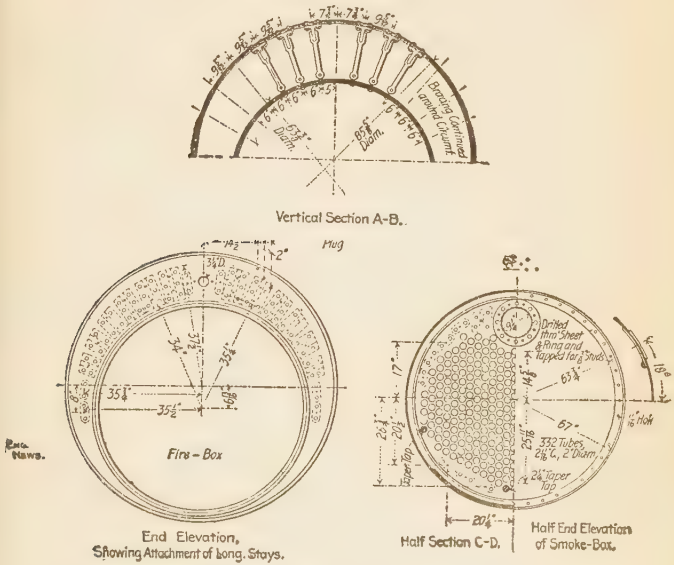


FIGURE 105.

be placed far enough apart to allow a man to pass between them for the purposes of inspection and washing out of the boiler. Through stay rods are usually spaced 14 in. apart horizontally and about the same distance vertically. The ends, as far back as the threads run, are swaged larger than the body, so that the diameter at the bottom of the thread is greater than the diameter of the body. There are several

methods of applying through stays. One of the most common, especially for land boilers, is to allow the ends of the rod to project through the plates to be stayed, and holding them in place by a nut and copper washer, both inside and outside the plate. Another and still better plan is to rivet 6-in. channel bars across the head, inside above the tubes, the number of bars depending upon the height of the segment to be stayed.

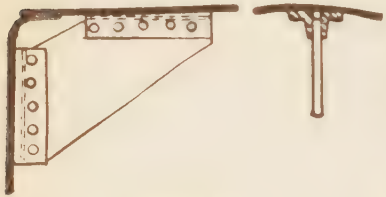


FIGURE 106.

The channel bars are drilled to correspond with the holes that are drilled in the plate to receive the stay rods, which latter are then secured by inside and outside nuts and copper washers.

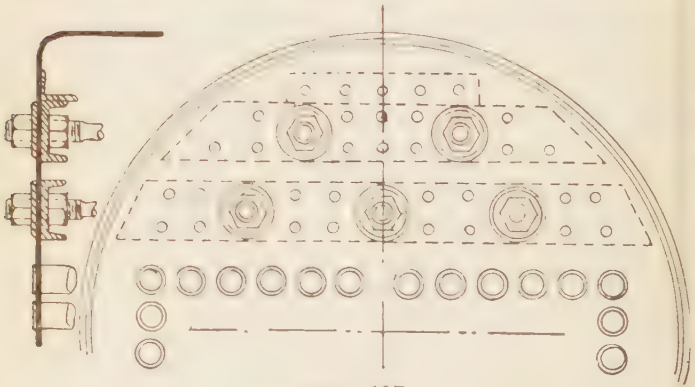


FIGURE 107.

These channel bars act as girders and serve to greatly strengthen the head or flat plate. Fig. 107 will serve to illustrate this method.

Sometimes a combination of channel bar and diagonal crow foot braces is used, as shown by Fig. 108.

A good form of diagonal crow foot stay is obtained by using double crow feet, made of pieces of boiler plate bent as shown by Fig. 109 and riveted to the plate by four rivets. A hole is drilled through the body of the crow foot, and a bolt passing through this secures the forked end of the stay.

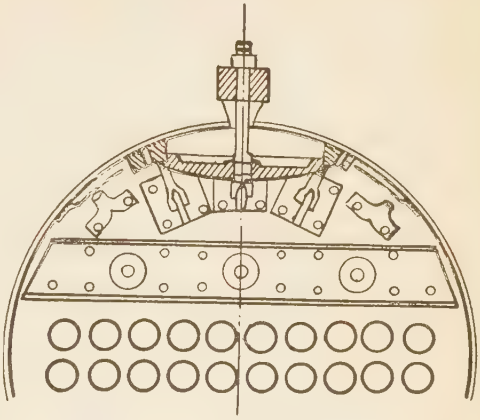


FIGURE 108.

Another method of securing through stays to the heads is shown by Fig. 110 and is applied where too many stay rods would be required to connect all the points to be stayed. A tee iron is first riveted to the flat plate to be stayed, and two V-shaped forgings are bolted to it as shown. The through stay is then bolted to the forgings, and thus two points in the flat head are supported by one stay. It will readily be seen that this method will reduce the number of through stay rods required.

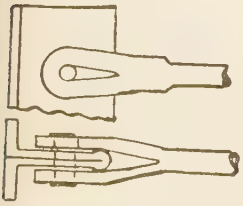


FIGURE 109.

Calculating the Strength of Stayed Surfaces. In calculations for ascertaining the strength of stayed sur-

faces, or for finding the number of stays required for any given flat surface in a boiler, the working pressure being known, it must be remembered that each stay is subjected to the pressure on an area bounded by lines drawn midway between it and its neighbors. Therefore the area in square inches, of the surface to

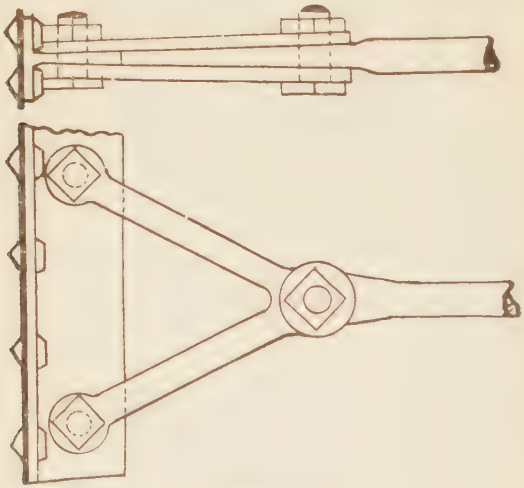


FIGURE 110.

be supported by each stay, equals the square of the pitch or distance in inches between centers of the points of connection of the stays to the flat plate. Thus, suppose the stays in a certain boiler are spaced 8 in. apart, the area sustained by each stay = $8 \times 8 = 64$ sq. in., or assume the stay bolts in a locomotive fire box to be pitched $4\frac{1}{2}$ in. each way, the area supported by each stay bolt = $4\frac{1}{2} \times 4\frac{1}{2} = 20\frac{1}{4}$ sq. in. Again taking through stay rods, suppose, for example,

the through stays shown in Fig. 107 to be spaced 15 in. horizontally and 14 in. vertically, the area supported by each stay = 15 × 14 = 210 sq. in.

The minimum factor of safety for stays, stay bolts and braces is 8, and this factor should enter into all computations of the strength of stayed surfaces.

The pitch for stays depends upon the thickness of the plate to be supported, and the maximum pressure to be carried.

In computing the total area of the stayed surface it is safe to assume that the flange of the plate, where it is riveted to the shell, sufficiently strengthens the plate for a distance of 2 in. from the shell, also that the tubes act as stays for a space of 2 in. above the top row. Therefore the area of that portion of the flat head or plate bounded by an imaginary line drawn at a distance of 2 in. from the shell and the same distance from the last row of tubes is the area to be stayed. This surface may be in the form of a segment of a circle, as with a horizontal cylindrical boiler, or it may be rectangular in shape, as in the case of a locomotive or other fire box boiler. Other forms of stayed surfaces are often encountered, but in general the rules applicable to segments or rectangular figures will suffice for ascertaining the areas.

The method of finding the area of the segmental portion of the head above the tubes is given in Chapter I, pages 22 and 23, and will not be enlarged upon here, except to add Table 19, which covers a much greater number of segments than Table 1, page 22, does. The diameter of the circle and the rise or height of the segment being known, the area of the segment may be found by the following rule:

Rule. Divide the height of the segment by the

TABLE 19

Areas of Segments of a Circle

Ratio	Area	Ratio	Area	Ratio	Area	Ratio	Area
.2	.11182	.243	.14751	.286	.18542	.329	.22509
.201	.11262	.244	.14837	.287	.18633	.33	.22603
.202	.11343	.245	.14923	.288	.18723	.331	.22697
.203	.11423	.246	.15009	.289	.18814	.332	.22792
.204	.11504	.247	.15095	.29	.18905	.333	.22886
.205	.11584	.248	.15182	.291	.18996	.334	.22980
.206	.11665	.249	.15268	.292	.19086	.335	.23074
.207	.11746	.25	.15355	.293	.19177	.336	.23169
.208	.11827	.251	.15441	.294	.19268	.337	.23263
.209	.11908	.252	.15528	.295	.19360	.338	.23358
.21	.11990	.253	.15615	.296	.19451	.339	.23453
.211	.12071	.254	.15702	.297	.19542	.34	.23547
.212	.12153	.255	.15789	.298	.19634	.341	.23642
.213	.12235	.256	.15876	.299	.19725	.342	.23737
.214	.12317	.257	.15964	.3	.19817	.343	.23832
.215	.12399	.258	.16051	.301	.19908	.344	.23927
.216	.12481	.259	.16139	.302	.20000	.345	.24022
.217	.12563	.26	.16226	.303	.20092	.346	.24117
.218	.12646	.261	.16314	.304	.20184	.347	.24212
.219	.12729	.262	.16402	.305	.20276	.348	.24307
.22	.12811	.263	.16490	.306	.20368	.349	.24403
.221	.12894	.264	.16578	.307	.20460	.35	.24498
.222	.12977	.265	.16666	.308	.20553	.351	.24593
.223	.13060	.266	.16755	.309	.20645	.352	.24689
.224	.13144	.267	.16843	.31	.20738	.353	.24784
.225	.13227	.268	.16932	.311	.20830	.354	.24880
.226	.13311	.269	.17020	.312	.20923	.355	.24976
.227	.13395	.27	.17109	.313	.21015	.356	.25071
.228	.13478	.271	.17198	.314	.21108	.357	.25167
.229	.13562	.272	.17287	.315	.21201	.358	.25263
.23	.13646	.273	.17376	.316	.21294	.359	.25359
.231	.13731	.274	.17465	.317	.21387	.36	.25455
.232	.13815	.275	.17554	.318	.21480	.361	.25551
.233	.13900	.276	.17644	.319	.21573	.362	.25647
.234	.13984	.277	.17733	.32	.21667	.363	.25743
.235	.14069	.278	.17823	.321	.21760	.364	.25839
.236	.14154	.279	.17912	.322	.21853	.365	.25936
.237	.14239	.280	.18002	.323	.21947	.366	.26032
.238	.14324	.281	.18092	.324	.22040	.367	.26128
.239	.14409	.282	.18182	.325	.22134	.368	.26225
.24	.14494	.283	.18272	.326	.22228	.369	.26321
.241	.14580	.284	.18362	.327	.22322	.37	.26418
.242	.14666	.285	.18452	.328	.22415	.371	.26514

TABLE 19—*Continued*

Ratio	Area	Ratio	Area	Ratio	Area	Ratio	Area
.372	.26611	.405	.29827	.438	.33086	.471	.36373
.373	.26708	.406	.29926	.439	.33185	.472	.36471
.374	.26805	.407	.30024	.44	.33284	.473	.36571
.375	.26901	.408	.30122	.441	.33384	.474	.36671
.376	.26998	.409	.30220	.442	.33483	.475	.36771
.377	.27095	.41	.30319	.443	.33582	.476	.26871
.378	.27192	.411	.30417	.444	.33682	.477	.36971
.379	.27289	.412	.30516	.445	.33781	.478	.37071
.38	.27386	.413	.30614	.446	.33880	.479	.37171
.381	.27483	.414	.30712	.447	.33980	.48	.37270
.382	.27580	.415	.30811	.448	.34079	.481	.37370
.383	.27678	.416	.30910	.449	.34179	.482	.37470
.384	.27775	.417	.31008	.45	.34278	.483	.37570
.385	.27872	.418	.31107	.451	.34378	.484	.37670
.386	.27969	.419	.31205	.452	.34477	.485	.37770
.387	.28067	.42	.31304	.453	.34577	.486	.37870
.388	.28164	.421	.31403	.454	.34676	.487	.37970
.389	.28262	.422	.31502	.455	.34776	.488	.38070
.39	.28359	.423	.31600	.456	.34876	.489	.38170
.391	.28457	.424	.31699	.457	.34975	.49	.38270
.392	.28554	.425	.31798	.458	.35075	.491	.38370
.393	.28652	.426	.31897	.459	.35175	.492	.38470
.394	.28750	.427	.31996	.46	.35274	.493	.38570
.395	.28848	.428	.32095	.461	.35374	.494	.38670
.396	.28945	.429	.32194	.462	.35474	.495	.38770
.397	.29043	.43	.32293	.463	.35573	.496	.38870
.398	.29141	.431	.32392	.464	.35673	.497	.38970
.399	.29239	.432	.32941	.465	.35773	.498	.39070
.4	.29337	.433	.32590	.466	.35873	.499	.39170
.401	.29435	.434	.32689	.467	.35972	.5	.39270
.402	.29533	.435	.32788	.468	.36072		
.403	.29631	.436	.32887	.469	.36172		
.404	.29729	.437	.32987	.47	.36272		

diameter of the circle. Then find the decimal opposite this ratio in the column headed "Area." Multiply this area by the square of the diameter. The result is the required area.

Example. Diameter of circle = 72 in. Height of segment = 25 in. $25 \div 72 = .347$, which will be found in the column headed "Ratio," and the area opposite this

is .24212. Then $.24212 \times 72 \times 72 = 1.255$ sq. in., area of segment.

A boiler is 66 in. in diameter, the working pressure is 100 lbs. per sq. in. The distance from the top row of tubes to the shell is 25 in. Required, the number of diagonal crow foot braces that will be needed to support the heads above the tubes, also the sectional area of each brace. The thickness of the heads is $\frac{5}{8}$ in. and the T.S. = 55,000 lbs. per sq. in.

Assume the head to be sufficiently strengthened by the flange for a distance of 2 in. from the shell, the diameter of the circle of which the segment above the tubes requires to be stayed is reduced by $2 + 2 = 4$ in. and will therefore be $66 - 4 = 62$ in. The rise or height of the segment above the tubes is $25 - 4 = 21$ in. Required, the area.* $21 \div 62 = .338$. Looking down the column headed "Ratio" in Table 19, area opposite .338 is .23358. Area of segment $= .23358 \times 62 \times 62 = 897.88$ sq. in. The total pressure on this area will be $897.88 \times 100 = 89{,}788$ lbs.

Assume the braces to be made of $1\frac{1}{8}$ in. round steel having a T.S. of 50,000 lbs. per sq. in. and to be designed in such a manner as to allow for loss of material in drilling the rivet holes in the crow feet. Each brace will have a sectional area of .994 sq. in., and using 8 as a factor of safety, the strength or safe holding power of each stay may be found as follows: $.994 \times 50{,}000 \div 8 = 6{,}212$ lbs., and the number of stays required = 89,788 lbs. (total pressure) divided by 6,212 lbs. (strength of each stay) = 14.5, or in round numbers 15. If the stays are made of flat bars of steel the sectional area should equal that of the round stays and the dimensions of the crow feet of all stays should

*See rule for Table 19.

be such as to retain the full sectional area of the body after the rivet holes are drilled.

Each stay is connected to the plate by two ⅞-in. rivets, having a T.S. of 55,000 lbs. per sq. in. and a shearing strength of 45,000 lbs. per sq. in. These rivets are capable of resisting a direct pull of 10,818 lbs., using 5 as a factor of safety; ascertained as follows: $2A \times 45{,}000 \div 5 = 10{,}818$ = strength of two rivets. They are also subjected to a crushing strain, and the resistance to this is $D \times C \div 5$, which expressed in figures is $.875 \times 90{,}000 \div 5 = 15{,}750$ lbs.

The proper spacing comes next, and is arrived at in the following manner:

Area to be stayed = 897.88 sq. in.

Number of stays = 15.

Area supported by each stay = 897.88 ÷ 15 = 59.8 sq. in.

The square root of 59.8 = 7.75 nearly, which is the distance in inches each way that the stays should be spaced, center to center.

If through stay rods are used in place of diagonal braces for staying the boiler under consideration, the number and diameter of the rods may be ascertained by the following method:

Assuming the heads to be supported by channel bars, as previously described, and that the stays are pitched 14 in. apart horizontally and 13 in. vertically, each stay would be required to support an area of $14 \times 13 = 182$ sq. in., and the number of stays would be $897.88 \div 182 = 4.9$, in round numbers 5. See Fig. 107. The pressure being 100 lbs. per sq. in., the total stress on each stay = $182 \times 100 = 18{,}200$ lbs. Assume the stay rods to be of soft steel having a T.S. of 50,000 lbs. per sq. in., and using a factor of safety of 8, the

sectional area required for each stay will be found as follows: 18,200 × 8 ÷ 50,000 = 2.9 sq. in., and the diameter will be found as follows: 2.9 ÷ .7854 = 3.69. which is the square of the diameter, and the square root of 3.69 = 1.9 in., or practically 2 in. The same methods of calculation are applicable to the staying of the heads below the tubes, also for stay bolts in fire box boilers.

Strength of Unstayed Surfaces. A simple rule for finding the bursting pressure of unstayed flat surfaces is that of Mr. Nichols, published in the "Locomotive," February, 1890, and quoted by Prof. Kent in his "Pocket-book." The rule is as follows: "Multiply the thickness of the plate in inches by ten times the tensile strength of the material used, and divide the product by the area of the head in square inches." Thus,

Diameter of head = 66 in.
Thickness of head = 5/8 in.
Tensile strength = 55,000 lbs.
Area of head = 3,421 sq. in.

5/8 × 55,000 × 10 ÷ 3,421 = 100, which is the number of pounds pressure per square inch under which the unstayed head would bulge.

If we use a factor of safety of 8, the safe working pressure would be 100 ÷ 8 = 12.5 lbs. per sq. in., but as the strength of the unstayed head is at best an uncertain quantity it has not been considered in the foregoing calculations for bracing, except as regards that portion of it that is strengthened by the flange.

In all calculations for the strength of stayed surfaces, and especially where diagonal crow foot stays are used, the strength of the rivets connecting the stay to the flat plate must be carefully considered. A large factor of safety, never less than 8, should be used, and

the cross section of that portion of the foot of the stay through which the rivet holes are drilled should be large enough, after deducting the diameter of the hole, to equal the sectional area of the body of the stay.

Dished Heads. In boiler work where it is possible to use dished, or "bumped up" heads as they are sometimes called, this type of head is rapidly coming into use. Dished heads may be used in the construction of steam drums, also in many cases for dome-covers, thus obviating the necessity of bracing. The maximum depth of dish, as adopted by steel plate manufacturers April 4, 1901, is ⅛ of the diameter of the head when flanged, and if the tensile strength and quality of the plate from which the heads are made are the same as those of the shell plate, the dished head becomes as strong as the shell, provided the head has the same thickness or is slightly thicker than the shell plate.

Welded Seams. A few boiler manufacturers have succeeded in making welded seams, thus dispensing with the time-honored custom of riveting the plates together. A good welded joint approaches more nearly to the full strength of the material than can possibly be attained by rivets, no matter how correctly designed the riveted joint may be. The weld also dispenses with the necessity of caulking, and a boiler having a perfectly smooth surface inside, such as would be afforded by welded seams, would certainly be much less liable to collect scale and sediment than would one with riveted joints. But in order to make a success of welded seams the material used must be of the best possible quality, and great care and skill are required in the work.

The Continental Iron Works of Brooklyn, New York, exhibited at the St. Louis World's Fair in 1904

a welded steel plate soda pulp digester without a single riveted joint. The dimensions of this vessel, which may be likened to a cylinder boiler without flues, were as follows: Thickness of plate, $\frac{1}{2}$ in.; diameter, 9 ft.; length, 43 feet. The heads were dished to the standard depth. The safe working pressure was 125 lbs. per sq. in. It appears not only possible, but probable, that the process of welding boiler joints may in time supplant the older custom of riveting.

CHAPTER II

CARE OF THE BOILER

Washing out the boiler—Duties of the boiler washer—How to prepare a boiler for washing—How to clean and inspect the inside of a boiler—Fusible plugs—Advantage of manholes, giving free access to top and bottom of boiler—Responsibility resting upon the boiler washer—Necessity of keeping water column clean—Scraping the flues—Fire cracks and how to deal with them—Firing up and how it should be done—Danger in too sudden heating up of a boiler—Advantages of filling a recently washed out boiler with warm water—Connecting with the main header and the safest method of procedure.

Washing Out. In order to get the best results from the burning of coal or any other fuel in a boiler furnace it is absolutely necessary to keep the boiler as clean as possible, both inside and outside. In large plants the boiler washer and his helper are detailed to look after this part of the work, and while the job is by no means a very desirable one, it is at the same time a very responsible one, and much depends upon the thoroughness with which the work is done. In small plants, consisting of one or two boilers, the engineer generally has to attend to the details of the work himself, and no matter whether the plant be large or small, the engineer in charge is the man above all others who should be most interested in seeing that thorough work is done, not only as a matter of safety, but for the sake of his reputation as an engineer. The boiler that is to be washed out should be allowed to gradually cool for ten or twelve hours. It will then be in a condition which will permit a man to go inside of it

and do effective work, and no boiler can be thoroughly cleaned and inspected unless the boiler washer does go inside.

These remarks apply, of course, to horizontal tubular or flue boilers and water tube boilers having drums large enough for a man to crawl into. Some types of internally fired boilers are provided with man-holes, but the majority of them have only hand-holes into which the hose for washing out may be inserted.

After the water has been allowed to run out, the first step in washing out a boiler is to remove all the loose mud and scale possible by means of a steel scraper fitted to a long handle and introduced through the man-hole in the bottom part of the head. This will prevent the scale from getting into the blow-off pipe and stopping the flow of the water used for washing the boiler. If there is a man-hole on top, the next thing in order is to take the hose in through it and give the sides of the shell and also the tubes a good cleaning.

Sometimes it happens that where an exhaust heater of the open type is used, oil will find its way into the boiler and, mixing with the mud, will form a thick pasty-like substance on the sides of the boiler along the water line. This should be carefully scraped off and removed, as any matter containing oil or grease is a very dangerous thing to have inside a boiler.

After cleaning the upper part of the boiler, it should be inspected for loose braces or rivets. This can best be accomplished by tapping the parts with a light hammer. A solid rivet will give a clear metallic sound, and a little practice will enable one to easily detect the sound of a loose brace or broken rivet.

Signs of corrosion or pitting of the shell along the

water line should also be carefully searched for. Fusible plugs, to be effective, must be kept clean, and the only opportunity for cleaning them is at the time of washing out the boiler. Therefore while working on the upper part of the boiler, attention should be given to the fusible plug. If it is one of the ordinary kind, screwed into the back head above the tubes, it should be taken out and cleaned and before replacing it the thread should be well coated with a mixture of cylinder oil and plumbago, which will prevent it from sticking. If the fusible plug is one of the type consisting of a brass tube extending from the top of the shell to the water level, the lower end of this tube should be cleaned of all mud or scale.

Having thus finished above the tubes, the mud and scale should again be scraped from the bottom, after which the hose should be inserted through the front man-hole that should be in every horizontal boiler.

Some authorities argue that a man-hole should not be cut in the bottom part of a boiler head, giving as their reason that it weakens the head, but the logic is not sound, for the reason that the man-hole can be reënforced in such a manner as to make it fully as strong as the solid sheet, and when we consider the great advantage of having a man-hole in the bottom, both as regards washing out and also for repairs, it is plain that it is really a necessity.

After washing out all the loose mud and scale that it is possible to get from the bottom, the boiler washer should next go inside and, with scrapers and tools made for the purpose, he should scrape and chip off all the scale that he can from the bottom, because there is where lies the greatest danger from burnt sheets caused by accumulations of scale preventing the

water from getting to the metal. Much good work may be accomplished in this way and no boiler washer should consider the job complete until he has gone through the boiler both top and bottom, and not only cleaned but inspected it. Any loose rivets, broken or loose braces, signs of corrosion or pitting should be at once reported to the chief engineer or superintendent.

It will thus be seen that great responsibility rests with the boiler washer, for the reason that he is the man that is in closest touch with the inside of the boiler, and it is due to the manner in which he does his work inside the boiler whether a defect is discovered and repaired in time or whether it is allowed to go until the result is often a grave disaster. The author desires to enter a plea for this hard-worked and too often underpaid craftsman, and hereby expresses the wish that his services were better appreciated.

The water column or combination should receive particular attention each time the boiler is washed out. The lower pipe leading to the boiler is liable to become clogged with scale, and if not cleaned regularly it is sure to cause trouble by preventing a free flow of the water from the boiler to the gauge glass.

If the boiler is of the horizontal tubular type, the tubes should be scraped inside, and with water tube boilers use the steam jet to blow the soot and ashes from between the tubes. Soot, in addition to choking the draft, is also a non-conductor of heat.

After the hand-hole and man-hole plates have been replaced the boiler may be filled with water to the proper level, and while this is being done it is in order to take a look into the furnace for any broken grates or accumulations of clinkers on the side walls or bridge wall. These clinkers should be chipped off, also the

bottom of the boiler should be swept clean of ashes and examined for any defects, such as fire cracks about the rivets most exposed to the heat. These cracks may often exist some time before being discovered unless a close inspection is made. They are small cracks radiating from the rivet holes outward past the rivet heads one-half to three-quarters of an inch, and are always liable to extend farther until they become a source of danger unless arrested in time. They may be closed up sometimes with the caulking tool, but if one should be found several inches in length, a hole should be drilled at the outer end of the crack and a rivet put in. This will generally stop it. Fire cracks occur in the girth seams only, and especially the seam nearest the fire.

It is essential that the bridge walls of horizontal boilers be kept in good repair, in order that as much fire brick surface as possible may be exposed to the heat. This will greatly aid combustion and prevent smoke.

Firing Up. After the boiler washer has completed his task the next thing in order is firing up, and in doing this if care and good judgment are not exercised there is danger of doing much damage to the boiler, especially if it has been filled with cold water. A very light fire should be started at first and kept that way until the water gets to the boiling point at least, after which the fire may be gradually increased until the steam gauge shows a few pounds pressure, when it will be safe to urge the fire still more. The bad effects resulting from the unequal expansion or contraction of the sheets and undue stress upon the rivets, all caused by rapid changes of the temperature of the boiler from hot to cold or vice versa, cannot be

guarded against too carefully, and they are liable to be brought about in two ways: first, by haste in cooling down a hot boiler that is to be washed out, and secondly, by starting a heavy fire under a cold boiler. That part of the boiler most exposed to the heat will become hot while other parts farthest removed from the fire may still be cold. Very often there is a difference of 150° or 200° in the temperature of different parts of the boiler for a time during the firing up process, and the same dangerous conditions may be caused also by blowing all the water out of a boiler while under a pressure of 15 or 20 lbs., as is the custom of some persons when preparing a boiler for washing out. Either custom cannot be too strongly condemned.

Sometimes a boiler is needed in a hurry after having been washed out, and in such an emergency it should be filled with warm water; in fact, it is better to always fill a boiler with warm water if it is possible to do so after washing out.

Connecting with the Main Header. When the gauge shows a pressure that is within 10 or 15 lbs. of being the same as that carried on the other boilers it should be watched closely, and when the pressure becomes the same as that in the main the connecting valve should be opened slightly, just sufficient to allow a light flow of steam through it, which can be easily detected by placing the ear near the valve chamber. This steam may be passing from the boiler to the header or vice versa, but whichever way it is going the valve should not be opened any farther until the pressure in the main pipe and in the boiler is equalized, when it will be found that the valve may be opened easily. While connecting the boiler the dampers should be closed.

Care should always be exercised in connecting a recently fired up boiler, and the engineer should be certain that the steam gauge and pop valve are in good working order. Otherwise there is liability of a serious accident occurring, either in breakage of the steam pipe, or what is still worse, a boiler explosion.

Steam Boilers

CHAPTER III

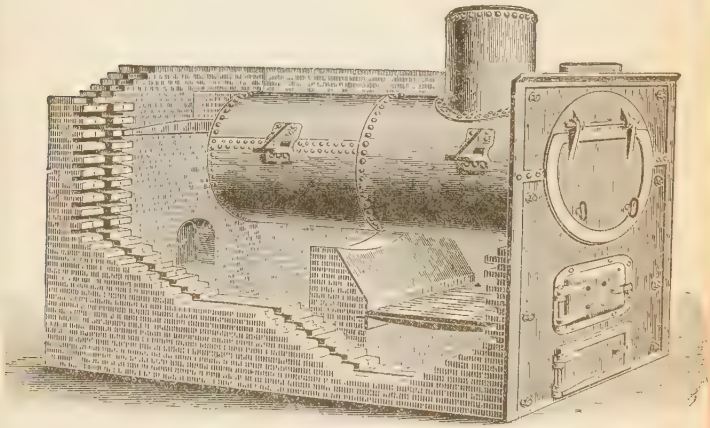
THE BOILER

Description of various types—Construction—Rules for ascertaining strength of sheet before and after punching—Strength of rivets—Single and double riveted seams—Triple riveted butt joints, strength of—Force tending to rupture a boiler—Rules for finding the safe working pressure of boilers—Bracing—Rules for bracing—Bracing domes.

It is hardly within the scope of this book to describe the many and varied types of metallic vessels known as steam boilers in use to-day for the generation of steam for power and other purposes. The author will deal mainly with those types most commonly used in this country for stationary service.

Description. These may be divided into four different classes. The first and most simple type, and the one from which the others have gradually evolved, is the plain cylinder boiler in which the heated gases merely pass under the boiler, coming in contact only with the lower half of the shell and then pass to the stack. These boilers are generally of small diameter (about 30 in.) and great length (30 ft.). Next comes the flue cylindrical boiler, which is somewhat larger in diameter than the former, generally 40 in. diameter and 20 to 30 ft. long, with two large flues 12 to 14 in. diameter extending through it. The return tubular boiler, consisting of a shell with tubes of small diam-

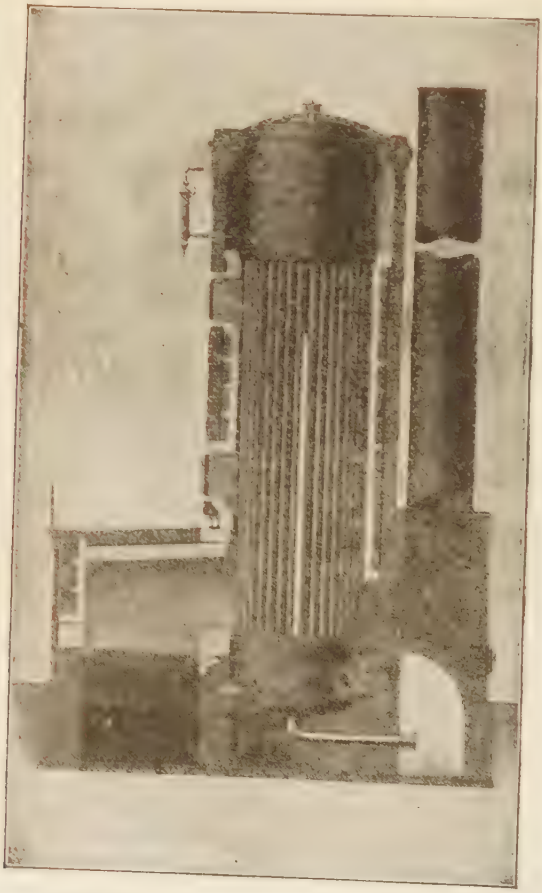
eter (2 to 4 in.) extending from head to head through which the hot gases from the furnace pass on their way to the stack. This boiler, which comes in the third class, is probably more extensively used in the United States for stationary service than any other type. The fourth class comprises the water tube boilers, in which the water is carried in tubes 3 to 4 in. in diameter, sometimes vertical and sometimes inclined, and connected at the top to one end of a steam drum,



STANDARD HORIZONTAL BOILER WITH FULL-ARCH FRONT SETTING.

and having the lower ends of the tubes connected to a mud drum, which is also connected to the opposite end of the steam drum, thus providing for a free circulation of the water. Of the latter type there have been many different kinds evolved during the last one hundred years, the majority of them having had but a brief existence, being compelled to obey the inexorable law of the survival of the fittest, and to-day there are a few excellent types of water tube boilers

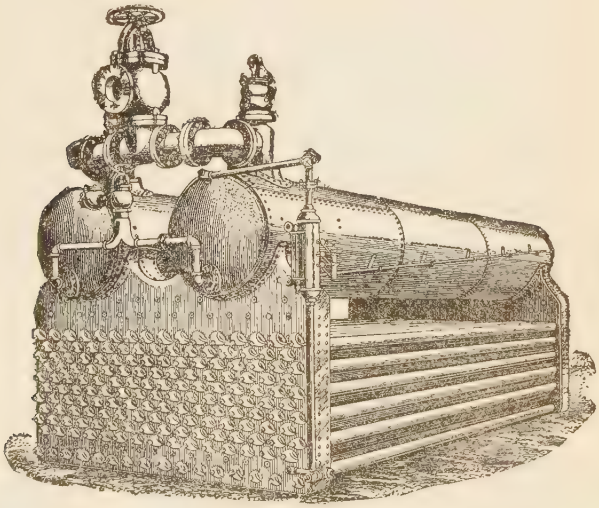
which have become standard and are being extensively used. The margin of safety as regards disastrous



WICKES VERTICAL WATER TUBE BOILER.

explosions appears to be in favor of the water tube boiler. It is not contended that they are entirely

exempt from the danger of explosion. On the contrary, the percentage of explosions of water tube boilers in proportion to the number in use is probably as large, if not larger, than it is with boilers of the shell or return tubular type, but the results are seldom so destructive of life or property, for the reason that if one or more of the tubes give way the pressure is released and the danger is past.



500 HORSE POWER HEINE WATER TUBE BOILER.

Construction. As the four classes of boilers above referred to are constructed of similar material, although assembled in different ways, the standard rules for calculating strength of joints, bracing, etc., may be applied to all.

The shell should be made of homogeneous steel of about 60,000 lbs. tensile strength. The thickness depending upon the pressure to be carried. The term

tensile strength means that it would take a pull of 60,000 lbs. in the direction of its length to break a bar of the material one inch square, or two inches wide by one-half inch thick, or three-eighths of an inch thick by 2.67 in. wide.

The heads are generally made one-eighth of an inch thicker than the shell.

Riveting. Boiler rivets should be of good charcoal iron, or a soft, mild steel of 38,000 lbs. to 40,000 lbs. T. S. No boiler is stronger than its weakest part, and it is evident that a riveted joint has not the full strength of the solid plate. In order to ascertain the safe working pressure of a boiler it is necessary to first determine the strength of the riveted seams, and the method of doing this is as follows: Assume the boiler to be of the horizontal tubular type, 60 in. in diameter by 16 ft. in length. The plates to be of steel $\frac{3}{8}$ in. thick, having a tensile strength of 60,000 lbs. per square inch, the longitudinal seams to be double riveted and the girth seams to be single riveted. The pitch of the rivets, that is the distance from the center of one rivet hole to the center of the next one in the same row, to be for the double riveted seams $3\frac{1}{4}$ in. and for the single riveted seams $2\frac{1}{2}$ in. The diameter of the rivets to be $\frac{3}{4}$ in. and diameter of holes to be $\frac{13}{16}$ in. Assume the rivets to have a T. S. of 38,000 lbs. per square inch of sectional area. First, find strength of a section of solid plate $3\frac{1}{4}$ in. wide, which is the width between centers of rivet holes before punching.

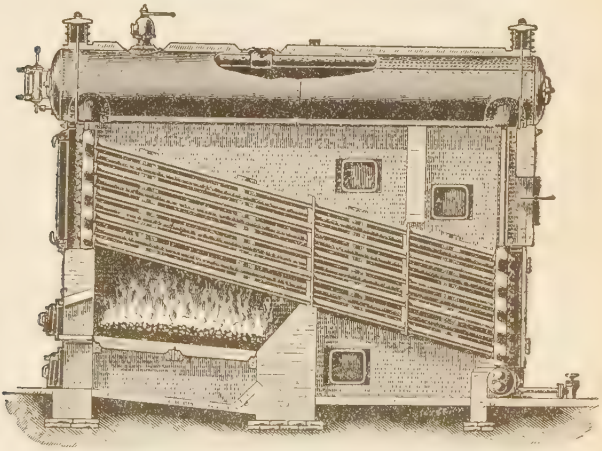
Rule 1. Pitch × thickness × T. S. Thus, $3.25 \times .375 \times 60{,}000 = 73{,}125$ lbs., strength of solid plate.

Second, find strength of net section of plate, meaning that portion of plate left after deducting the diam-

eter of one hole $\frac{15}{16}$ in., which expressed in decimals = .9375 in. from the width of plate before punching.

Rule 2. Pitch – diameter of hole × thickness × T. S. Thus, 3.25 – .9375 × .375 × 60,000 = 52,031 lbs., strength of net section of plate.

Third, find strength of rivets. In calculating the strength of rivets in a double riveted seam, the sectional area of two rivets must be considered, taking

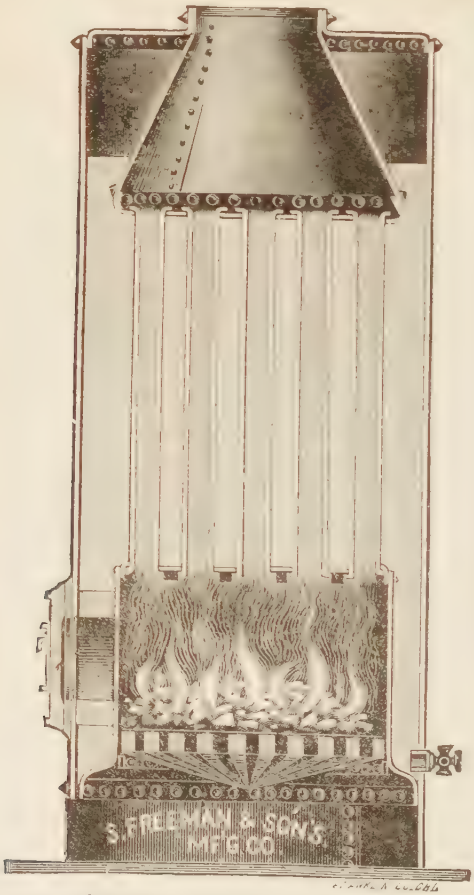


BABCOCK AND WILCOX BOILER.

one-half the area of two rivets in the first row, and the area of another rivet in the second row. The area of a $\frac{7}{8}$-in. rivet is .6013 in., but when in position it is assumed to fill the hole $\frac{15}{16}$ in. Consequently, its area would then be .69 in. and its strength is found by Rule 3.

Rule 3. Sectional area × T. S. Thus, .69 × 38,000 = 26,220 lbs., strength of one rivet, and multiplying by 2, as there are two rivets, the result is 26,220 × 2 = 52,440

lbs., strength of rivets in the seam under consideration. It thus appears that the plate is the weakest por-



VERTICAL FIRE BOX BOILER.

tion and the percentage of strength retained is found by multiplying 52,031 by 100 and dividing by 73,125

the strength of solid plate. Thus, $\frac{52,031 \times 100}{73,125} = 71.1$ per cent.

The query might arise, why is the diameter of one rivet hole deducted from the pitch when figuring the strength of net plate? The answer is, that in punching the holes one-half the diameter of each hole is cut from the section designated, thereby reducing its width by just that amount.

The 71.1 per cent. obtained by the calculation represents the strength of the boiler as compared to the strength of the sheet before punching, and should enter into all calculations for the safe working pressure.

It is usual in practice to figure the strength of a double-riveted seam at 70 per cent. of the strength of the solid plate. The strength of triple-riveted butt joints may be calculated by taking a section of plate along the first row of rivets and estimate it as a single-riveted joint, then add to this result the strength of rivets in the second and third rows for a section of the same width. In properly designed triple-riveted butt joints the percentage of strength retained is 88, and some recent achievements in designing have shown the remarkable result of quadruple-riveted butt joints retaining as high as 92 to 94 per cent. of the strength of the solid plate.

Bursting Pressure. The query might arise, why should the longitudinal or side seams require to be stronger than the girth or round about seams? The answer is, that the force tending to rupture the boiler along the line of the longitudinal seams is proportional to the diameter divided by two, while the stress tending to pull it apart endwise is only one-half that, or proportional to the diameter divided by four.

To illustrate, let Fig. 1 represent the shell of the boiler heretofore referred to, ignoring for the time being the tubes and braces, and consider the boiler simply as a hollow cylinder. Now the total force tending to rupture the boiler along the line of the girth seams or in the direction of the horizontal arrows = area of one head in square inches × pressure in pounds per square inch. It is true that the pressure is exerted against both heads, but the area of one head can only be considered for the reason that the two stresses are exerted against each other just as in the case of two horses pulling against each other, or in opposite direction on the same chain. The stress on the chain will

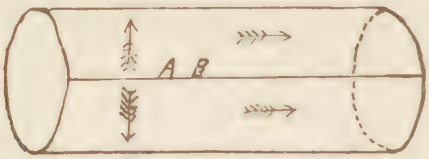


FIGURE 1.

be what each horse (not both) pulls. To further illustrate, suppose one of the horses to be replaced by a permanent post or wall and let one end of the chain be attached thereto. One head or one side of the boiler pulls against the other, and the stress on the seams is the force with which each (not both) pulls. Referring again to Fig. 1, area of one head = $60^2 \times .7854 = 2827.4$ sq. in. Suppose there is a pressure of 10 lbs. per square inch in the boiler. Then total stress on the girth seams = $2827.4 \times 10 = 28,274$ lbs. Opposed to this pull is the entire circumference of the boiler, which is $60 \times 3.1416 = 188.5$ in. Therefore, dividing total pressure (28,274 lbs.) by the circumference in inches (188.5) will give 150 lbs. as the stress on each inch of the

girth seams. While the stress on each inch of the longitudinal seams or along the line A B, Fig. 1, and which is exerted in the direction of the vertical arrows, is pressure (10 lbs.) × one-half the diameter (30 in.) = 300 lbs. One-half the diameter is used because the pressure in any direction is effective only on the surface at right angles to that direction.

The formula for finding the bursting pressure of a boiler may be expressed as follows:

$B = \frac{T.S. \times T}{R}$ in which B = bursting pressure.
T.S. = tensile strength.
T = thickness of sheet.
R = radius or one-half the diam.

Example. T. S. = 55,000 lbs. per square inch.
T = ⅜ in. (expressed decimally = .375 in.).
R = 30 in.

Then 55,000 × .375 ÷ 30 = 687.5 lbs. per square inch, which is the pressure at which rupture would take place provided there were no seams in the boiler and the original strength of the sheet was retained, but, as has been seen, a certain percentage of strength is lost through punching or drilling the necessary rivet holes, and this must be taken into account.

The formula now becomes, for double riveting, $B = \frac{T.S. \times T \times .70}{R}$, in which the letters preserve the same value as in the original formula, but the result is reduced by multiplying by the decimal .70, which represents the percentage of strength retained by double-riveted seams. Consequently B will now =

$$\frac{55{,}000 \times .375 \times .70}{30} = 481 \text{ lbs.}$$

In case the seams are all single riveted .56 must be

substituted for .70, and with triple-riveted butt joints .88 can safely be used.

Safe Working Pressure. In order to ascertain the safe working pressure of a boiler it is necessary first to calculate the bursting pressure and divide this by another factor called the factor of safety. The one most commonly used for boilers is 5, or in other words the safe working pressure = one-fifth the bursting pressure. In the case of the boiler under consideration, the safe pressure would be 481 ÷ 5 = 96 lbs., at which point the safety valve should blow off.

Bracing. Every engineer can easily ascertain for himself whether the boilers under his charge are properly braced or not. The parts that require bracing are: all flat surfaces, such as the sides and top of the fire-box in boilers of the locomotive type, and those portions of the heads above and below the tubes in horizontal tubular boilers, also the top of the dome.

The stress per square inch of sectional area on braces and stays should not exceed 6,000 lbs. It is customary to consider the flange of the head and the top row of tubes as sufficient bracing for a space two inches wide above the tubes and the same distance around the flange. Therefore the part of the head to be braced will be the segment contained within a line drawn two inches above the top row of tubes and two inches inside the flange.

In order to ascertain the number of braces required for a given boiler head, three factors are necessary: first, the area of the segment in square inches; second, the diameter and T. S. of the braces, and third, the pressure to be carried. By the use of Table No. 1 the areas of segments of boiler heads ranging from 42 to 72 in. in diameter can easily be obtained. Assume the

boiler to be 60 in. in diameter, distance from top of tubes to top of shell 24 in. Deduct 4 in. for surface braced by top row of tubes and flange, leaving the height of segment to be braced 20 in.

TABLE I

Diameter of Boiler.	Ditsance from Tubes to Shell.	Height of Segment.	Constant.
42 in.	15 in.	11 in.	.16314
44 in.	17 in.	13 in.	.1936
48 in.	19 in.	15 in.	.20923
54 in.	21 in.	17 in.	.21201
60 in.	24 in.	20 in.	.22886
66 in.	25 in.	21 in.	.214
72 in.	29 in.	25 in.	.24212

Rule. Multiply the square of the diameter of the boiler by the constant number found in right hand column opposite column headed diameter.

Example. $60 \times 60 \times .22886 = 823.89$ sq. in., area of segment to be braced. Find number of braces required. Assume the braces to be 1⅛ in. in diameter and of a T. S. of 38,000 lbs. per square inch of section. The area of one brace will be .994 sq. in., which × 6,000 lbs. gives 5,964 lbs. as the stress allowable on each brace. Suppose the pressure to be carried is 100 lbs. per square inch. There will be area of segment (823.89 sq. in.) × pressure (100 lbs.) = 82,389 lbs., total stress. Dividing this result by 5,964 lbs. (the capacity of each brace) gives 13.8 braces as the number needed. In practice there should be fourteen.

Having a T. S. of 38,000 lbs. and using 6 as the factor of safety, each brace could safely sustain a pull of 6,295 lbs. Therefore it is evident that the above mentioned load for each brace is well within the limit. For convenience in calculating the areas of segments

of circles, other than those mentioned in Table 1, the following rule is given:

Referring to Figure 2 it is desired to find the area of the segment contained within the lines A B C E. It will be necessary first to find the area of the sector bounded by the lines A B C D. This is done by multiplying one-half the length of the arc, A B C, by the radius, D B. Having obtained the area of the sector, the next step is to find the area of the triangle bounded by the lines A E C D and subtract it from the area of the sector. The remainder will be the area of the segment. Having found the area of the surface to be braced, and the number of braces required, it now becomes necessary to consider the spacing of the same.

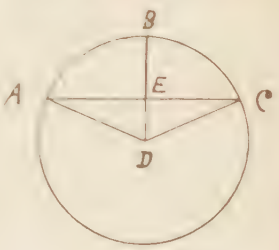


FIGURE 2.

Rule. Divide area to be braced by the number of braces, and extract the square root of quotient.

Example. 823.80 ÷ 14 = 58.8 sq. in. to be allotted to each brace. Extract square root of 58.8 and the result is 7.68 inches, which is the length of one side of the square which each brace will be required to sustain.

For internally fired boilers the same rules can be applied except that the surfaces to be braced are generally of rectangular shape and consequently the area is more easily figured than in the case of segments. That part of the head below the tubes also requires to be braced, and two braces are generally sufficient, as at A and B, Fig 3. In the case of domes it is safe to consider the portion of the head within

three inches of the flange as sufficiently braced. Then suppose the dome to be 36 in. in diameter, there will remain a circle 30 in. in diameter to be braced. The circumference of this circle is 94.2 in. and the pitch, or distance from center to center of the braces, being 7.6 in., the number of braces required is found by dividing 94.2 by 7.6, giving 12 braces. These braces should be located along a line which is one-half the pitch, or 3.8 in., within the circumference of the 30-in. circle. The space immediately surrounding the hole cut for the steam outlet will be sufficiently reënforced by the flange riveted on for the reception of the steam pipe. All holes cut in boilers, such as man holes, hand holes, and those for pipe connections, above two inches should be properly reënforced by riveting either inside or outside a wrought-iron or steel ring or flange of such thickness and width as to contain at least as much material as has been cut from the hole.

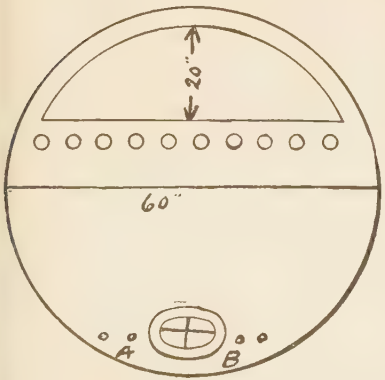


FIGURE 3.

Questions

1. How many types of steam boilers are there?
2. What kind of boilers are included in type one?
3. Describe a boiler belonging to type two.
4. Describe a boiler of the third type.

5. How is a boiler of the fourth type constructed?

6. In what respect do water tube boilers have the advantage over other types as regards explosions?

7. What kind of material should be used in the construction of boilers?

8. What does the term tensile strength (T. S.) mean?

9. What is the usual T. S. of steel boiler plates?

10. How much thicker than the shell plates should the heads be?

11. Of what material and of what T. S. should the rivets be?

12. Is a riveted joint as strong as the solid plate?

13. What is meant by the pitch of the rivets?

14. What is the usual pitch for a double riveted seam?

15. Give the rule for finding the strength of the solid plate before punching.

16. What is meant by net section of plate?

17. What is the rule for finding strength of net section of plate?

18. How is the strength of rivets in a double riveted seam calculated?

19. What percentage of the strength of solid plate is usually retained in a double riveted seam?

20. How is the strength of a triple riveted butt strap joint calculated?

21. What per cent. of the original strength of the sheet is retained in a properly designed triple riveted butt strap joint?

22. Why should the side seams be stronger than the girth seams?

23. What is the rule for finding the bursting pressure of a boiler?

24. How is the safe working pressure of a boiler calculated?

25. What parts of a boiler require bracing internally?

26. What stress per square inch of sectional area may be allowed on braces?

27. How is the number of braces required for any part of the boiler obtained?

28. How should domes be braced?

ANSWERS

1. Four.

2. Plain cylinder boilers without tubes or flues. The hot gases merely pass under, and come in contact with, the lower half of the shell and then escape to the stack.

3. It is termed a cylindrical flue boiler. It is usually about 40 inches in diameter and 20 to 30 feet in length, having two large flues 12 to 14 inches in diameter extending through it from head to head.

4. It is termed a return tubular boiler, and consists of a shell having tubes of small diameter (2 to 4 inches) located in the water space, and extending from head to head, through which the hot gases from the furnace pass on their way to the stack.

5. It is termed a water tube boiler and consists of a set, or sets of tubes, 3 to 4 inches in diameter, in some boilers vertical and in others inclined and connected at the top to a steam drum, and at the bottom to a mud drum. The water circulates freely through these tubes.

6. The margin of safety regarding disastrous explosions appears to be in their favor.

7. The shell should be made of homogeneous steel, of about 60,000 lbs. tensile strength, the thickness depending upon the pressure to be carried.

8. The number of pounds pull in the direction of its length that would be required to break a bar of the material one inch square, or two inches in width by one-half inch thick, or three-eighths of an inch thick by 2.67 inches wide.

9. 60,000 lbs. per sq. in.

10. About one-eighth of an inch.

11. Good charcoal iron, or a soft mild steel of 38,000 lbs. to 40,000 lbs. T. S.

12. It is not.

13. The distance in inches from the center of one rivet hole to the center of the next one in the same row.

14. 3¼ inches.

15. Multiply thickness of plate by pitch of rivets, and this product by T. S.

16. That portion of the plate left after deducting the diameter of one rivet hole from the pitch.

17. Subtract diameter of one rivet hole from the pitch, and multiply the remainder by thickness of plate and by T. S.

18. Multiply sectional area of one rivet by its T. S., and this product by 2.

19. About 70 per cent.

20. Take a section of plate along the first row of rivets and estimate it as a single riveted joint, then add to this result the strength of rivets in the second and third rows for a section of the same width.

21. About 88 per cent.

22. For the reason that the force tending to rupture the boiler along the line of the side seams is proportional

to the diameter divided by 2; while the stress tending to pull it apart endwise is only one-half that, or proportional to the diameter divided by 4.

23. Multiply T. S. of sheet by its thickness and divide by one-half the diameter of boiler. Then for single riveted seams the bursting pressure will be 56 per cent of the result thus found, and for double riveted seams it will be 70 per cent, while for triple riveted butt joints 88 per cent of the original result will equal the bursting pressure.

24. First calculate the bursting pressure, then divide this by 5. Result will be safe working pressure in lbs. per sq. in.

25. All flat surfaces, including those portions of the heads above and below the tubes in horizontal tubular boilers, also the top of the dome. For boilers of the locomotive type; the sides of fire box, also the crown sheet require to be stayed.

26. It should not exceed 6,000 lbs. per sq. in. of sectional area of each brace.

27. Calculate area in square inches of surface to be braced and multiply this result by pressure in lbs. per sq. in. to be carried. Result will be total stress to be sustained by braces. Next multiply sectional area in square inches of one brace by 6,000. Result will be stress to be allowed on each brace. Then divide total stress by stress allowed for each brace. Result will be number of braces required.

28. All that portion of the head, or top of dome between the hole cut for the steam outlet and inside the circumference of a circle within three inches of the flange, should be reinforced by stay rods extending diagonally from the sides to the top of the dome.

CHAPTER IV

BOILER SETTINGS AND APPURTENANCES

Foundations—Brick work, etc.—Grate surface—Insulation—Water columns—Steam gages—Safety valves—Rules for finding areas of—Fusible plugs and where to place them—Domes and mud drums—Feed pipes—A good arch for the back connection—Blow off pipes and cocks—Surface blow off and circulating system—Feed pumps and feed water heaters—Injectors—Saving effected by heating the feed water with exhaust steam—Apparatus for making coal tests—Heating surface—Rules for figuring the same.

Settings. In the case of internally fired boilers the matter of setting resolves itself into the simple point of securing a sufficiently solid foundation, either of stone or brick laid in cement, for the boiler to rest upon.

But with horizontal tubular or water tube boilers the matter of brick work becomes important, and particular attention should be paid to securing a good foundation for the walls and great care exercised in building them in such manner that the expansion of the inner wall or lining will not seriously affect the outer walls. This can be done be leaving an air space of two inches in the rear and side walls, beginning at or near the level of the grate-bars and extending as high as the fire line, or about the center line of the boiler. Above this height the wall should be solid. Fig. 4 shows a plan and an end elevation illustrating this idea. The ends of some of the bricks should be allowed to project at intervals from the outer walls across the air space so as to come in touch with the inner walls.

Where boilers are set in batteries of two or more the middle or party walls should be built up solid from the foundation. All parts of the walls with which the

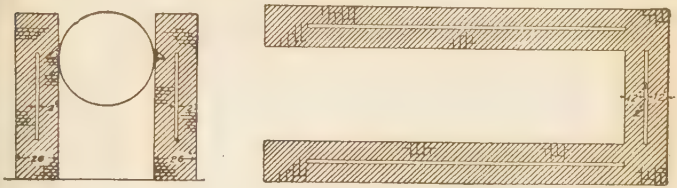


FIGURE 4.

fire comes in contact should be lined with fire brick, every fifth course being a header to tie the lining to the main wall.

Bridge walls should be built straight across from wall to wall of the setting, and should not be curved to conform to the circle of the boiler shell. The proper

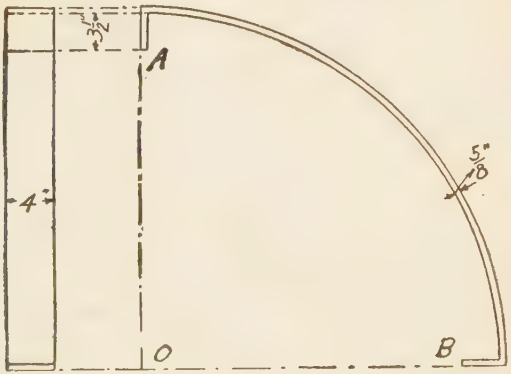


FIGURE 5.

distance from the top of the bridge wall to the bottom of the boiler varies from eight to ten inches, depending upon the size of the boiler. The space back of the

bridge wall, called the combustion chamber, can be filled in with earth or sand, and should slope gradually downward from the back of the bridge wall to the floor level at the rear wall, and should be paved with hard burned brick. The ashes and soot can then be easily cleaned out by means of a long-handled hoe or scraper inserted through the cleaning out door, which should always be placed in the back wall of every boiler setting.

Back Arches. A good and durable arch can be made

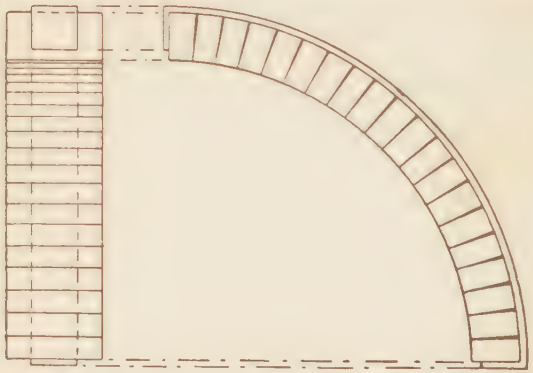


FIGURE 6.

for the back connection, extending from the back wall to the boiler head, by taking flat bars of iron $\frac{5}{8} \times 4$ in., cutting them to the proper length and bending them in the shape of an arch, turning four inches of each end back at right angles, as shown in Fig. 5. The distance O-B should equal that from the rear wall to the boiler head, and the height, O-A, should be about equal to O-B, and should bring the point A about two inches above the top row of tubes. The clamp thus formed is filled with a course of side arch fire brick,

Fig. 6, and will form a complete and self-sustaining arch, the bottom, B, resting on the back wall, and the top, A, supported by an angle iron riveted across the boiler head about three inches above the top row of tubes. See Figs. 7 and 8.

Enough of these arches should be made so that when laid side by side they will cover the distance from one side wall to the other across the rear end of the boiler. A fifty-four-inch boiler would thus require six clamps,

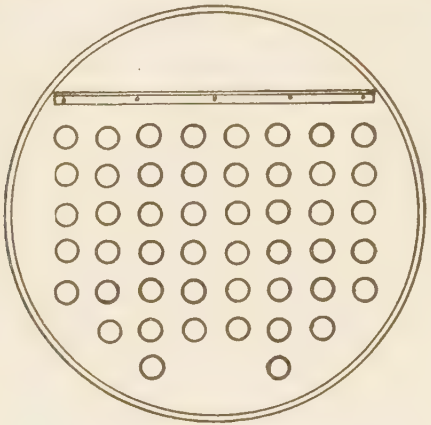


FIGURE 7.

a sixty-inch boiler seven clamps, and a seventy-two-inch boiler would require eight clamps; the length of a fire brick being about nine inches. In case of needed repairs to the back end of the boiler the sections can be lifted off, thus giving free access to all parts, and when the repairs are completed the arches can be reset with very little trouble and much less expense than the building of a solid arch would necessitate. This form of segmental arch allows ample freedom for expansion

of the boiler, in the direction of its length, without leaving an opening when the boiler contracts.

The crosswise construction of arch bars, while affording equal facility in repair work, is necessarily more expensive than the form here described, and is also open to the objection that it cannot follow the contracting boiler and maintain a tight joint or connection

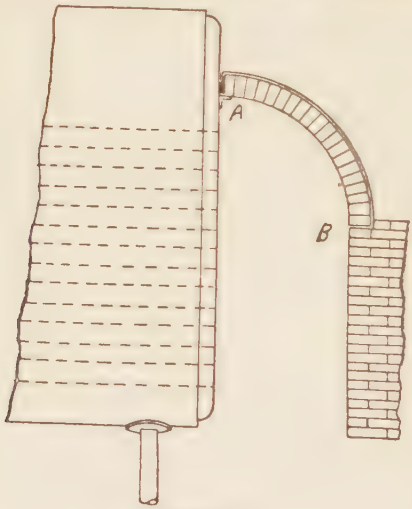


FIGURE 8.

between the back arch and the rear head above the tubes.

Boiler walls should always be well secured in both directions by tie rods extending throughout the entire length and breadth of the setting, whether there be one boiler or a battery of several. The bottom rods should be laid in place at the floor level when starting the brick work, and the top rods extending transversely across the boilers can be laid on top of the boilers.

The top rods extending from front to back can be laid in the side walls or rest on top of them. All tie rods should be at least one inch in diameter, and for batteries of several boilers they should be larger. The rods should extend three or four inches beyond the brick work, with good threads and nuts on each end to receive the buck stays. In laying down the transverse tie rods they should be located so as to allow the buck stays to bind the brick work where the greatest concentration of heat occurs.

Horizontal boilers should always be set at least one inch lower at the back end than at the front, to make sure that the rear ends of the tubes will be covered with water so long as any appears in the gauge glass, provided of course that the lower end of the glass is properly located with reference to the top row of tubes, which will be discussed later on. Upon the brick work and immediately under each lug of the boiler there should be laid in mortar a wrought or cast iron plate several inches larger in dimension than the bearing surface of the lug and not less than one inch in thickness. Upon each of these plates there should be placed two rollers made of round iron 1 or 1⅛ in. in diameter, and as long as the width of the lug. These rollers should be placed at right angles to the length of the boiler, in such a position that the lug will bear equally upon them. The object of the rollers is to prevent disturbance of the brick work by the endwise expansion and contraction of the boiler.

Grate Surface. The number of square feet of grate surface required depends upon the size of the boiler. A good rule and one easy to remember is to make the length of the grates equal to the diameter of the boiler. The width, of course, will depend upon the construc-

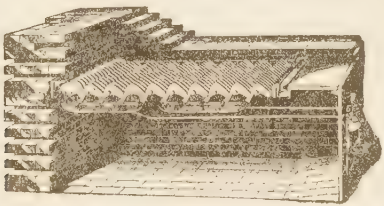
tion of the furnace. If the fire brick lining is built perpendicular, the width of grate will be about equal to the diameter of the boiler. On the other hand, if the lining is given a batter of three inches, starting at the level of the grate, then the width will be reduced six inches. It is customary to allow one square foot of grate surface to every 30 sq. ft. of heating surface. The distance of the grate-bars from the shell of the boiler varies from 24 to 28 in., according to the dimensions of the boiler.

Insulation. All boilers should be well protected from the cooling influence of outside air, if economy

M'CLAVE'S GRATES.

of fuel is any object. The tops of horizontal boilers should be covered with some kind of heat insulating material, or arched over with common brick, leaving a space of two inches between the boiler and the arch. The resulting saving in fuel will far more than compensate for the extra expense in a very short time. All cracks in the side and rear walls should be carefully pointed up with mortar or fire clay. One source of heat loss in return flue boilers is short circuiting from the furnace to the breeching, caused by the arches over the fire doors becoming loose and shaky, and allowing considerable of the heat to escape directly to the stack instead of passing under the boiler and

through the tubes. Another bad air leak often occurs at the back connection when the arch rests wholly upon iron bars imbedded in the side walls. This leak, as has already been noted, is caused by the expansion of the boiler, which gradually pushes the arch away from the back head until, in the course of time, there will be a space of 5/8 in. and sometimes 3/4 in. between the head and the arch. The obvious remedy for this is an arch that will go and come with the movement of the boiler, and such an arch can be secured by building it in sections, as illustrated by Fig. 3, and then riveting a piece of angle iron to the boiler head, above

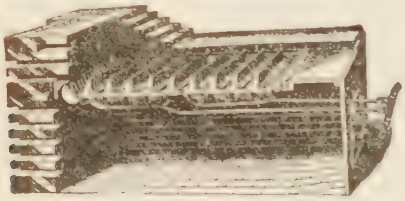


M'CLAVE'S GRATES.

the top row of tubes for the upper ends of the sections to rest upon, as already described. It will be seen that within all possible range of boiler movement in either direction the arch will, with this construction, always remain close to the head.

Water Columns. Water columns should be so located as to bring the lower end of the gauge glass exactly on a level with the top of the upper row of tubes, thus always affording a perfect guide as to the depth of water over the tubes. Many gauge glasses are placed too low, and water tenders and firemen are often deceived by them unless their positions with relation to the tubes are carefully noted.

The only safe plan for an engineer to pursue in taking charge of a steam plant is to seize the first opportunity for noting this relation. When he has washed out his boilers he may leave the top man-hole plates out while refilling them, and when the water stands at about four inches over the top row of tubes, the depth of water in the glass should be measured. He should do this with every boiler in the plant, and make a memorandum for each boiler. He will then know his bearings with regard to the safe height of water to be carried in the several gauge glasses. If he finds any of them are too low, he should lose no time

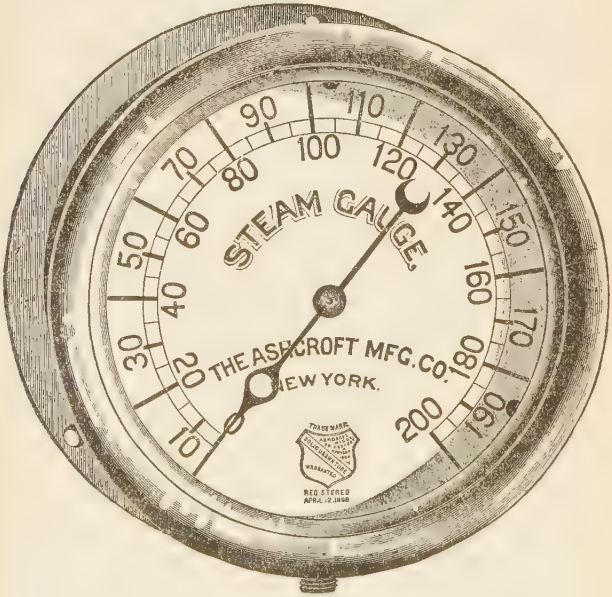


M'CLAVE'S GRATES.

in having them altered to comform to the requirements of safety. The position of the lower gauge cock should be three inches above the top row of tubes.

In making connections for the water column plugged crosses should always be used in place of ells. Brass plugs are to be preferred if they can be obtained; but whether of brass or iron, they should always be well coated with a paste made of graphite and cylinder oil before they are screwed in. They can then be easily removed when washing out the boiler, so as to allow the scale, which is sure to form in the lower connection, to be cleaned out. The best point at which to connect the lower pipe with the boiler is in the lower part of the

head just below the bottom row of tubes, and near the side of the boiler on which the water column is to stand; 1¼ or 1½ in. pipe should be used in all cases. The top connection can be made either in the head near the top, or in the shell. A ¾ or 1 in. drain pipe should be led into the ash pit, fitted with a good reli-

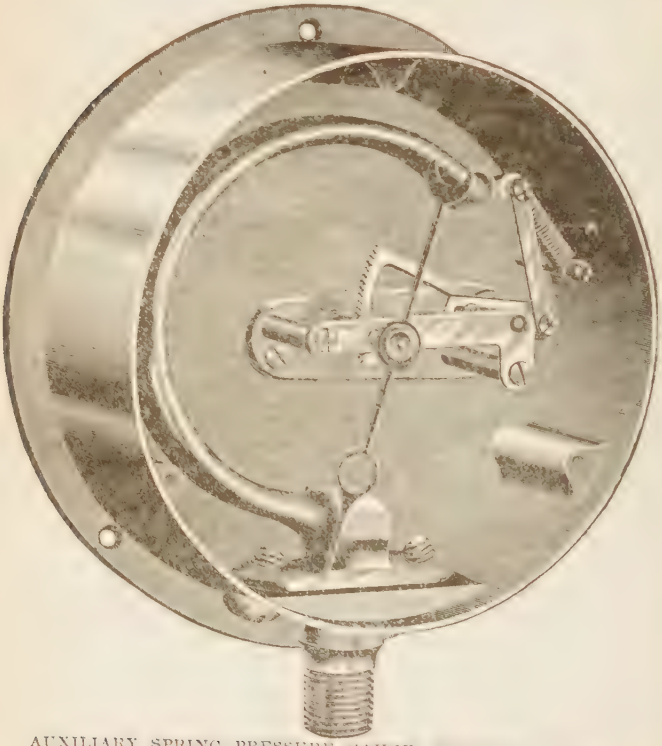


AUXILIARY SPRING PRESSURE GAUGE.

able valve which should be opened at frequent intervals to allow the mud and dirt to blow out of the water column and its connections. This is a very important point, and great care should be taken to keep the water column and all its connections thoroughly clean at all times.

One of the best indications that some portion of the

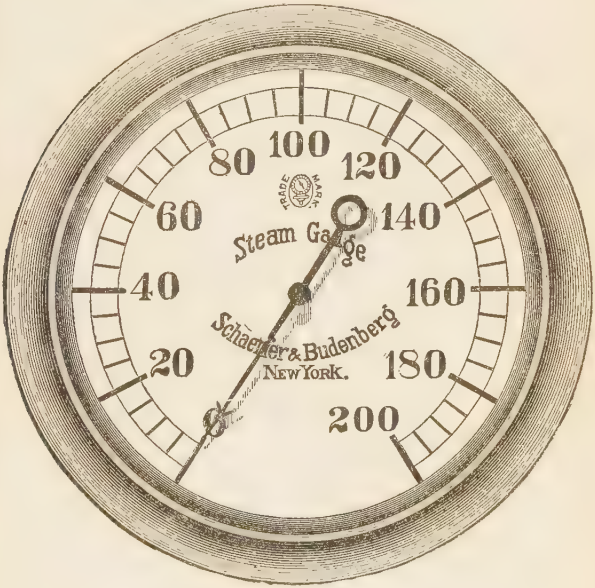
connections between the water glass and the boiler is choked or plugged with scale, is when there is no perceptible movement of the water in the glass. When the connections are free and the boiler is being fired,



AUXILIARY SPRING PRESSURE GAUGE, SECTIONAL VIEW.

there is always a slight movement of the water up and down in the glass, and when there is no perceptible movement it is time to look for the cause at once. Many instances of burned tubes have occurred, and

even explosions caused by low water in boilers while the gauge glass showed the water to be at a safe height. But owing to the connections having become plugged with scale, the water in the glass had no connection whatever with that in the boiler, and the water column was therefore worse than useless.



Steam Gauges. As water columns are made at present the steam gauge is usually connected at the top of the column. This makes a handsome and convenient connection, although theoretically the proper method would be to connect the steam gauge directly with the dome or the steam space of the shell. There should always be a trap or siphon in the gauge pipe in order

to retain the water of condensation, so as to prevent the hot steam from coming in contact with the spring

If at any time the water is drained from the siphon care should be exercised in turning on the steam again by allowing it to flow in very slowly at first until the siphon is again filled with water.

The steam gauge and the safety valve should be compared frequently by raising the steam pressure high enough to cause the valve to open at the point for which it is set to blow.

Safety Valves. The modern pop valve is generally reliable, but, like everything else, if it is allowed to stand idle too long it is likely to become rusty and stick. Therefore it should be allowed to blow off at least once or twice a week in order to keep it in good condition.

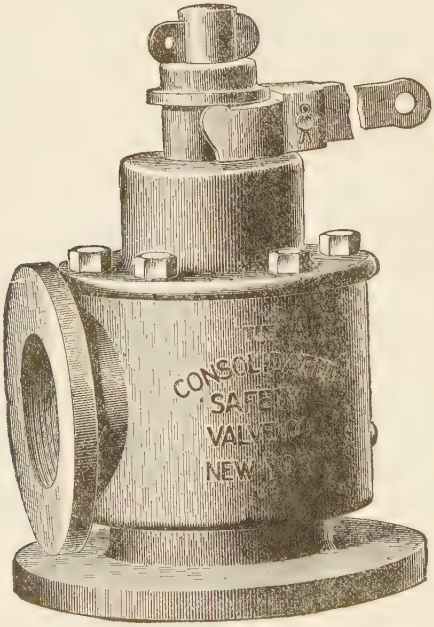
Most pop valves for stationary boilers are provided with a short lever, and if at any time the valve does not pop when the steam gauge shows the pressure to be high enough, it can generally be started by a light blow on the lever with a hammer.

The ratio of safety valve area to that of grate surface is, for the old style lever and weight valve, 1 sq. in. of valve area for each 2 sq. ft. of grate surface, and for pop valves 1 sq. in. of valve area for each 3 sq. ft. of grate surface.

Each boiler in a battery should have its own safety valve, and, in fact, be entirely independent of its mates as regards safety appliances.

One example of safety valve computation will be given. Suppose the grate surface of a boiler is $5 \times 6 = 30$ sq. ft, what should be the diameter of the lever safety valve? The required area of the valve is $30 \div 2 = 15$ sq. in. Then $15 \div .7854 = 19$, which is the

square of the diameter of the valve. Extracting the square root of 19 gives 4.35 in. diameter of valve. In actual practice one 5 in. or two 3 in. lever safety valves would be required. If a pop valve is to be used the required area is $30 \div 3 = 10$ sq. in. Then $10 \div .7854 = 12.73$ = square of diameter of valve. Extract the

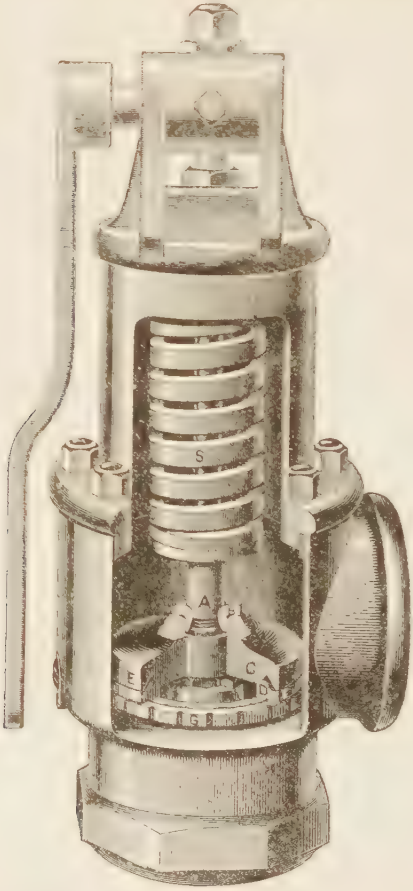


POP VALVE.

square root of 12.73 and the result is 3.6 in. = diameter of valve. In practice a 4 in. valve would be required.

Fusible Plugs. A fusible plug should be inserted in that part of the heating surface of a boiler which is first liable to be overheated from lack of water.

In a horizontal tubular or return flue boiler the proper location for the fusible plug is in the back head



INSIDE VIEW OF A POP SAFETY VALVE.

about 1½ or 2 in. above the top row of tubes. In fire-box boilers the plug can be put into the crown sheet

directly over the fire. These plugs should be made of brass with hexagon heads and standard pipe threads, in sizes ½, ¾, 1 in., or even larger if desired. A hole drilled axially through the center and counter sunk in the end that enters the boiler is filled with an alloy of such composition that it will melt and run out at the temperature of the dry steam at the pressure carried in the boiler. Thus, if the water should get below the plug the dry steam, coming in contact with the fusible alloy, melts it and, escaping through the hole in the plug, gives the alarm, and in case of fire-box or internally fired boilers the steam will generally extinguish the fire also. The hole is counter sunk on the inner end of the plug so as to retain the fusible metal against the boiler pressure. These plugs should be looked after each time the boilers are washed out, and all dirt and scale should be cleaned off in order that the fusible metal may be exposed to the heat.

Another type of fusible plug consists of a small brass cylinder into one end of which is screwed a plug filled with a metal which will fuse at the temperature of dry steam at the pressure which is to be carried in the boiler. The other end of the cylinder is reduced and fitted with a small stop valve and threaded to screw into a brass bushing inserted into the top of the boiler shell. This bushing also receives at its lower end a piece of ½ or ¾ in. pipe which extends downwards to within 2 in. of the top row of tubes, or the crown sheet if the boiler is internally fired. The principle of the device is that in case the water falls below the lower end of the pipe, steam will enter, fuse the metal in the plug, and be free to blow and give warning of danger. Some of these appliances are fitted with whistles which are sounded in case the steam gets access to

them. But even with such devices no engineer can afford to relax his own vigilance and depend entirely upon the safety appliances to prevent accidents from low water.

Domes and Mud-Drums. As a general proposition, both mud drums and domes are useless appendages to steam boilers. There are, no doubt, instances where they may serve a purpose, but as a rule their use is of no advantage to a boiler. Neither are the so-called circulating systems, sometimes attached to return tubular boilers, of any real value. These consist of one or more 4 to 6 in. pipes extending under the boiler from front to back through the furnace and the combustion chamber and connected to each end of the boiler.

Feed Pipes. Authorities differ in regard to the proper location of the inlet for the feed pipe, but upon one point all are agreed, namely, that the feed water, which is always at a lower temperature than the water in the boiler, should not be allowed to come directly in contact with the hot boiler sheets until its temperature has been raised to within a few degrees of the temperature of the water in the boiler. Certainly one of the most fruitful sources of leaks in the seams and around the rivets is the practice of introducing the feed water into the bottom either at the back or front ends of boilers, as is too often the case. The cool water coming directly in contact with the hot sheets causes alternate contraction and expansion, and results in leaks, and very often in small cracks in the sheet, the cracks extending radially from the rivet holes. It would appear that the proper method is to connect the feed pipe either into the front head just above the tubes, or into the top of the shell. The

nipple entering the boiler should have a long thread cut on the end which screws into the sheet, and to this end inside the boiler there should be connected another pipe which shall extend horizontally at least two-thirds of the length of the boiler, resting on top of the tubes, and then discharge. Or, what is still better, allow the internal pipe to extend from the entering nipple at the front end to within a few inches of the back head, then at right angles across the top of the tubes to the other side, and from there discharge downward. By this method the feed water is heated to nearly, if not quite, the temperature of the water in the boiler before it is discharged. One of the objections to this system is the liability of the pipe inside the boiler to become filled with scale and finally plugged entirely. In such cases the only remedy is to replace it with new pipe. But the great advantage of having the water thoroughly heated before being discharged into the boiler will much more than compensate for the extra expense of piping, and the general idea of introducing the feed water at the top instead of at the bottom of the boiler is therefore recommended as being the best.

The diameter of feed pipes ranges from 1 in. for small sized boilers, up to 1½ and 2 in. for boilers of 54 to 72 in. in diameter. It is not good policy to have the feed pipe larger than necessary for the capacity of the boiler; because it then acts as a sort of cooling reservoir for the feed water, and may cause considerable loss of heat.

For batteries of two or more boilers it is necessary to run a main feed header, with branch pipes leading to each boiler. The header should be large enough to supply all the boilers at the same time, should it ever

become necessary to do so. The header can be run along the front of the boilers just above the fire doors with the branch pipes running upon either side, clear of the flue doors and entering the front connection, or smoke arch, and the boiler head at a point two inches above the tubes. There should always be a valve in each branch pipe between the check valve and the header for the purpose of regulating the supply of water to each boiler, and also for shutting off the pump pressure in case of needed repairs to the check valve. Another valve should be placed between the check valve and the boiler. By this arrangement it is always possible to get at the check valve when it is out of order.

Blow off Pipes. Blow off pipes should always be connected with the lowest part of the water space of a boiler. If there is a mud-drum, then of course the blow off should be connected with it; but if there is no mud-drum, the blow off should connect with the bottom of the shell, near the back head, extend downwards to the floor of the combustion chamber, and thence horizontally out through the back wall, where the blow off cock can be located.

The best blow off cocks are the asbestos packed iron-body plug cocks, which are durable and safe. A globe valve should never be used in a blow off pipe, because the scale and dirt will lodge in it and prevent its being closed tightly. A straight way or gate valve is not so bad, but an asbestos packed plug cock is undoubtedly the best and safest.

In order to protect the blow off pipe from the intense heat, a shield consisting of a piece of larger pipe can be slipped over the vertical part before it is connected.

Blow off cocks should be opened for a few seconds once or twice a day, to allow the scale and mud to be blown out. If neglected too long they are liable to become filled with scale and burn out. A plan which is said to give good results is to connect a tee in the horizontal part of the pipe, and from this tee run a 1 in.

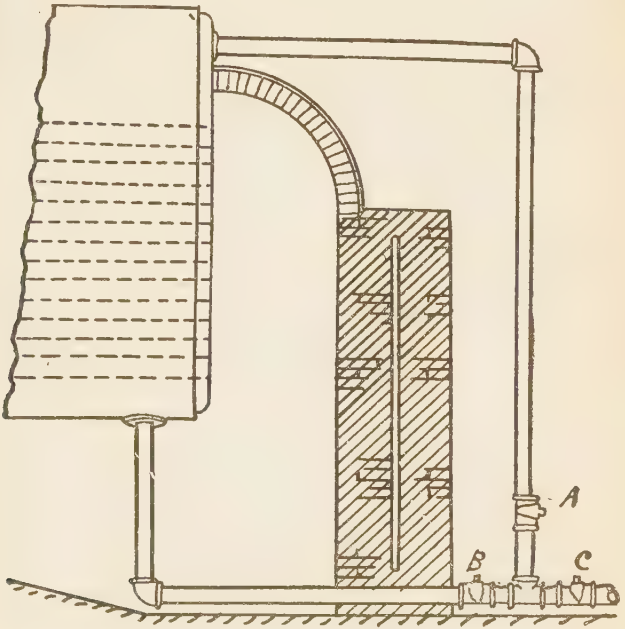


FIGURE 9.

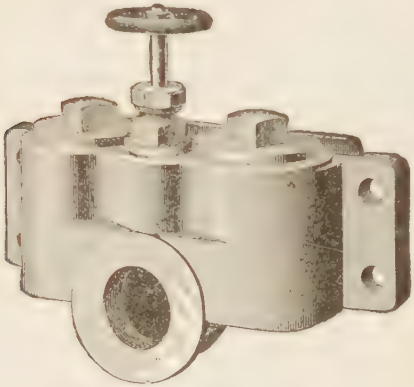
pipe to a point in the back head at the water level. It is claimed that this will cause a circulation of water in the pipe and prevent the formation of scale.

A surface blow off is a great advantage, especially if the water is muddy or liable to foam. By having the surface blow off connected on a level with the

water line a large amount of mud and other matter which is kept on the surface by the constant ebullition can be blown out.

A combination surface blow off, bottom blow off, and circulating system can be arranged by a connection such as illustrated in Fig. 6. By closing cock A and opening cocks B and C the bottom blow off is put in operation; by closing B and opening A and C the surface blow off is started, and by closing C and leaving A and B open the device will act as a circulating system. The pipe should be of the same size throughout. Blow off pipes should be of ample size, never less than 1¼ in., and from that to 2½ in., depending upon the size of the boiler.

Feed Pumps and Injectors. The belt driven power pump is the most economical boiler feeder, but is not

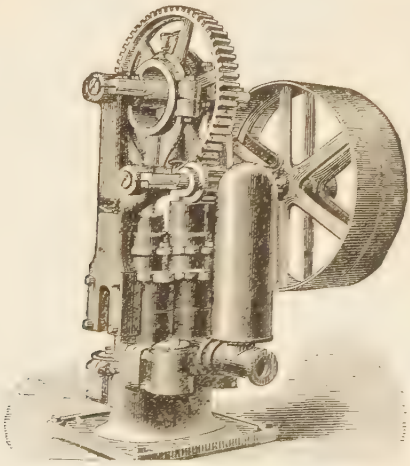


DIFFERENTIAL VALVE, DAVIS PUMP.

the most convenient nor the safest. When the engine stops, the pump stops also, and sometimes it happens that the belt gives way and the pump stops at just

the time when the boiler is being worked the hardest.

The modern double acting steam pump, of which there are many different makes to choose from, is without doubt the most reliable boiler feeding appliance and the one best adapted to all circumstances and conditions, although it is not economical in the

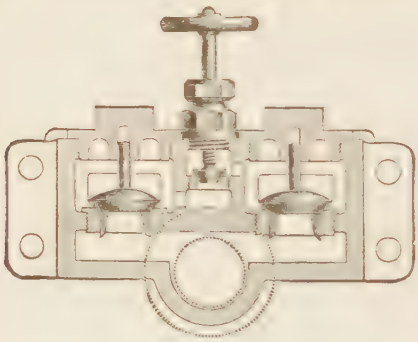


DAVIS BELT DRIVEN FEED PUMP.

use of steam, since the principle of expansion cannot be carried out with the pump as with the engine.

In selecting a feed pump care should be exercised to see that it is of the proper size and capacity to supply the maximum quantity of water that the boiler can evaporate. This may be ascertained by taking into consideration the amount of heating surface and the required consumption of coal per square foot of grate surface per hour. First, take the coal consumption. Assume the boiler to have 30 sq. ft. of grate surface,

and that it is desired to burn 15 lbs. of coal per square foot of grate per hour, which is a good average with the ordinary hand fired furnace using bituminous coal.



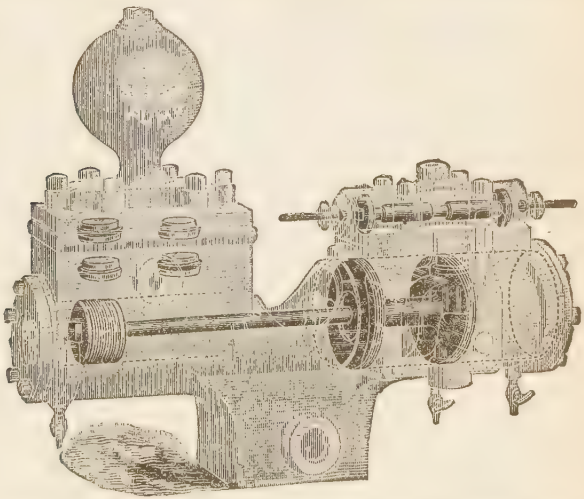
SECTIONAL VIEW OF DIFFERENTIAL VALVE.

Suppose the boiler is capable of evaporating 8 lbs. of water per pound of coal consumed. Then $30 \cdot 15 \times 8 =$ 3,600 lbs. of water evaporated per hour. Dividing

WORTHINGTON DUPLEX BOILER FEED PUMP.

3,600 by 62.4 (the weight of a cubic foot of water in pounds) gives 57.6 cu. ft. per hour, which, divided by 60, gives 0.96 cu. ft. per minute. This multiplied by

1,728 (number of cubic inches in a cubic foot) gives 1,659 cu. in per minute which the pump is required to supply. Suppose the pump is to make forty strokes per minute, and the length of stroke is five inches. Then 1659 ÷ 40 = 41.47 cu. in. per stroke, which, divided by 5 (length of stroke in inches) gives 8.294 sq. in. as the required area of water piston. 8.294 ÷ .7854 = 10.56, which is the square of the corresponding diam-



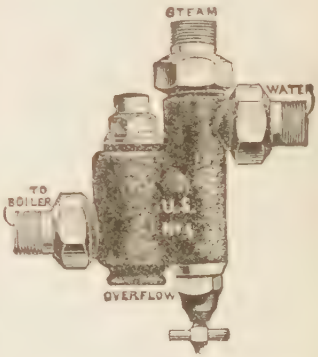
PHANTOM VIEW OF MARSH INDEPENDENT STEAM PUMP.

eter, and the square root of 10.56 = 3.25. So, theoretically, the size of the water end of the pump would be 3¼ in. in diameter by 5 in. stroke; but as it is always safer to have a reserve of pumping capacity, the proper size of the pump would be 3½ in. in diameter by 5 in. stroke, with a steam cylinder of 6 or 7 in. in diameter.

There is another rule for ascertaining the size of the

feed pump, by taking the number of square feet of heating surface in the boiler and allow a pump capacity of 1 cu. ft. per hour for each 15 sq. ft. of heating surface. Thus, let the total heating surface of the boiler be 786 sq. ft. Dividing this by 15 gives 52.4 as the number of cubic feet of water required per hour, from which the pump dimensions may be found in the same way as in the preceding case.

In figuring on the capacity of a feed pump for a battery of two or more boilers, the total quantity of water required by all the boilers must be taken into consideration. All boiler-rooms should be supplied with at least two feed pumps, so that if one breaks down there may always be another one available.

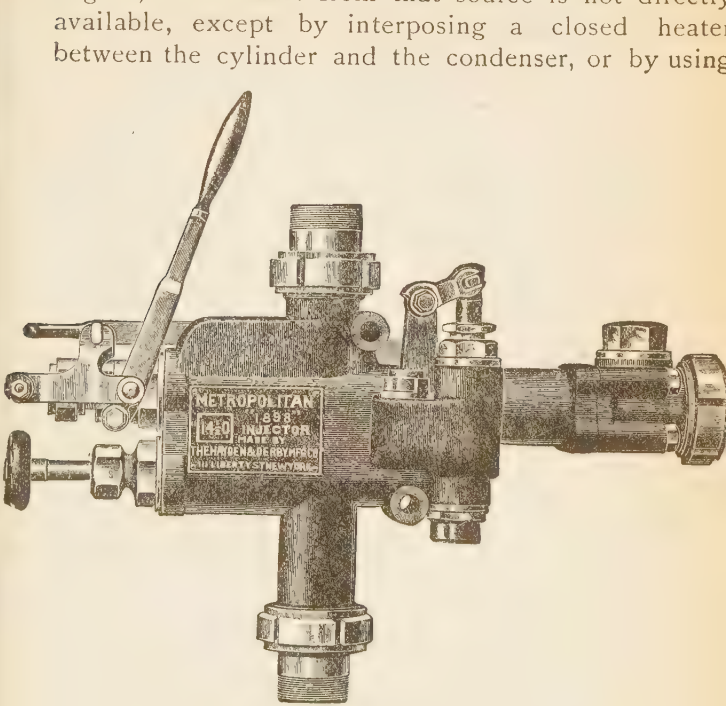


U. S. AUTOMATIC INJECTOR.

The injector is a reliable boiler feeder, and is in fact more economical than the steam pump, because the heat in the steam used is all returned to the boiler, excepting the losses by radiation. But the disadvantage attending the use of the injector is that it will not work well with the feed water at a temperature very much in excess of 100° F., while a good steam pump, fitted with hard rubber valves, will handle water at a temperature as high as 200° or 208° F., when the water flows to the pump by gravity from a heater, or it will raise water from a receiving tank on a short suction lift at a temperature of 150° or 160° F.

Feed Water Heaters. One great source of economy

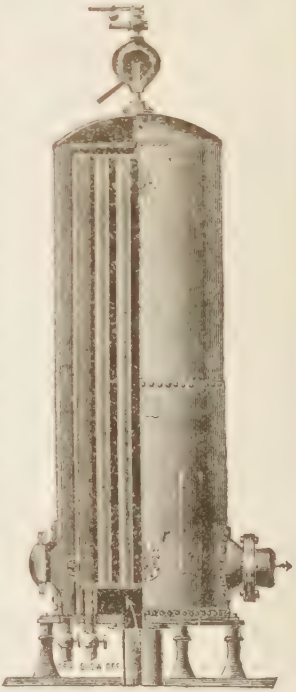
in fuel is the utilization of all the available exhaust steam for heating the feed water before it enters the boiler. Of course if the main engine is a condensing engine, the exhaust from that source is not directly available, except by interposing a closed heater between the cylinder and the condenser, or by using



METROPOLITAN INJECTOR, MODEL O.

the water of condensation for feeding the boilers. This can be done with safety, provided a surface condenser is used, but with a jet condenser or an open heater in which the exhaust mingles with the water, it is advisable to have an oil separator to prevent the oil from getting into the boilers.

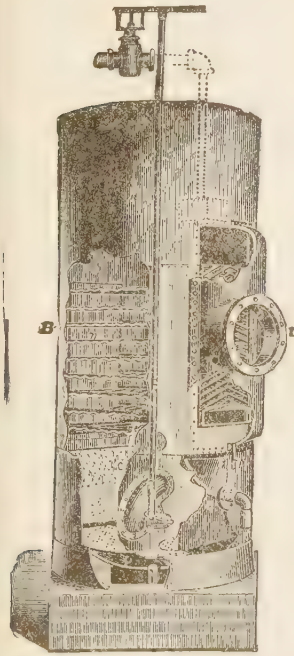
Exhaust heaters are of two kinds, open and closed. In the open heater the exhaust steam mingles directly with the water and a portion of it is condensed. A well-designed open exhaust heater will raise the temperature of the water to very nearly the boiling point, 212° F. These heaters should be set so that the water will flow by gravity from them to the feed pump In the closed type of exhaust heaters, the exhaust steam and the water are kept separate. In some styles the steam passes through tubes, which are surrounded by water, while in others the water fills the tubes, which are in turn surrounded by the steam. In either case the water in the closed heater is under the full boiler pressure while the feed pump is in operation, because the heater is between the pump and the boiler, while with the open heater the pump is between the heater and the boiler.



BARAGWANATH STEAM JACKET FEED WATER HEATER.

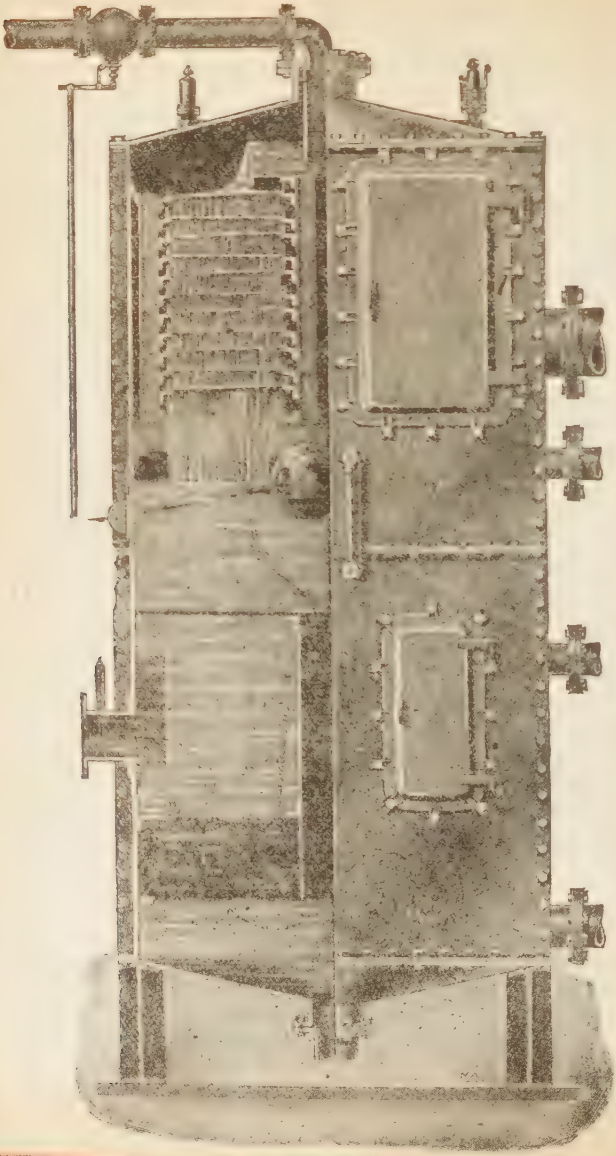
The saving effected by heating the feed water with exhaust steam can be easily ascertained by the use of a thermometer, a steam table, and a simple arithmetical calculation. First, find by thermometer the temperature of the water before entering the heater; find its temperature as it leaves the heater. Next ascertain by .

the steam table the number of heat units above 32° F. in the water at each of the two temperatures. Subtract the less from the greater, and the remainder will be the number of heat units added to the water by the heater. Next find by the table the number of heat units above 32° F. in the steam at the pressure ordinarily carried in the boiler, and subtract from this the number of heat units in the water before it enters the heater. The result will be the number of heat units that would be required to convert the water into steam of the required pressure, provided no heater were used. Then to find the percentage of saving effected by the heater, multiply the number of heat units added to the water by the heater by 100, and divide by the number of heat units required to convert the unheated water into steam, from the initial temperature at which it enters the heater.



INTERIOR VIEW OF OPEN HEATER.

Example. Assume the boiler to be carrying 100 lbs. gauge pressure. Suppose the temperature of the water before entering the heater is 60° F., and that after leaving the heater its temperature is 202° F., what is the percentage of saving due to the heater? The solution of the problem is as follows:



SQUARE OPEN HEATER.

Boiler pressure by gauge = 100 lbs.

Initial temperature of feed water = 60° F.

Heated temperature of feed water = 202° F.

From the steam table (see Chapter IV., Table 5) it is found that

Heat units in water at 202° F. = 170.7.

Heat units in water at 60° F. = 28.01.

Heat units added to water by heater = 170.7 – 28.01 = 142.69.

Heat units in steam at 100 lbs. gauge pressure = 1185.0.

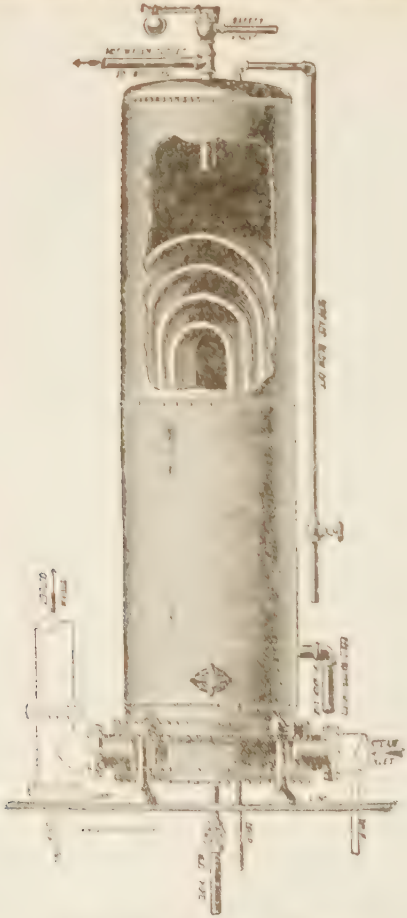
Heat units to be added to water at 60° F. to make steam of 100 lbs. gauge pressure = 1185.0 – 28.01 = 1156.99.

Percentage of saving effected by the use of the heater = $\frac{142.69 \times 100}{1156.99}$ = 12.33 per cent.

Suppose the coal consumed under this boiler amounts to two tons per day at a cost of $3.00 per ton, or a fuel cost of $6.00 per day. Then the saving in dollars and cents due to the heater in the foregoing example would be 12.33 per cent of $6.00, or $0.7398 (74 cents) per day.

Heaters, especially those of the closed type, should have capacity sufficient to supply the boiler for fifteen or twenty minutes. There would then be a body of water continually in the heater in direct contact with the heating surface, and as it passes slowly through it will receive much more heat than if rushed through a heater that is too small. All heaters and feed pipes should be well protected by some good insulating covering to prevent loss of heat by radiation. In some cases the exhaust steam, or a portion of it at least, can be used to advantage in an exhaust injector.

This device, where it can be used at all, is economical in that it not only feeds the boiler, but also heats the



CLOSED FEED WATER HEATER.

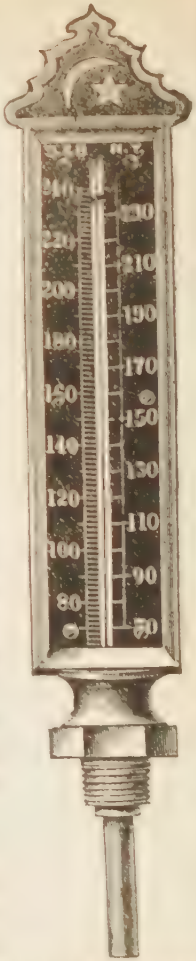
water without the use of live steam. But it will not force the water against a pressure much above 75 lbs.

to the square inch, and if the initial temperature of the water is much above 75° F. the exhaust injector will not handle it. Heaters which use live steam direct from the boilers heat the feed water to a much higher temperature, so that they act as purifiers by removing a large portion of the scale-forming impurities before the water enters the boiler. Live steam heaters, however, are not to be considered as economizers of heat.

Provisions for Testing. While considering feed pipes and other apparatus necessarily appertaining to the feeding of boilers, it is well to devote a short space also to the fittings and other devices required for successfully conducting tests of the boiler and furnace. This subject is mentioned here for the reason that the author considers that the necessary fittings and appliances for making evaporative tests properly belong to, and in fact are a part of, the feed piping, and can be put in while the plant is being erected at much less cost and trouble than if the matter is postponed until after the plant is in operation.

Beginning then at the check valve, there should be a tee located in the horizontal section of the feed pipe, as near to the check valve as practicable, and between it and the feed pump; or a tee can be used in place of an ell to connect the vertical and horizontal sections of the branch pipe where it rises in front of the boiler. One opening of this tee is reduced to ⅜ or ½ in. to permit the attachment of a hot water thermometer. These thermometers are also made angle-shaped at the shank, so that if desired they can be screwed into a tee placed in vertical pipe and still allow the scale to stand vertical. The thermometer is for the purpose of showing at what temperature the feed water enters

the boiler during the test, and therefore should be as near the boiler as possible. After the test is completed the thermometer may be taken out and a plug inserted in its place.



HOT WATER THERMOMETER.

The next requirement will be a device of some kind for ascertaining the weight of water pumped into the boiler during the test. In some well ordered plants each boiler is fitted with a hot water meter in the feed pipe, but as this arrangement is hardly within the reach of all, a substitute equally as accurate can be made by placing two small water tanks, each having a capacity of eight or ten cubic feet, in the vicinity of the feed pump. These tanks can be made of light tank iron, and each should be fitted with a nipple and valve near the bottom for connection with the suction side of the pump. The tops of the tanks may be left open If an open heater is used, and it is possible to place the tanks low enough to allow a portion of the water from the heater to be led into them by gravity, it will be desirable to do so. A pipe leading from the main water supply, with a branch to each tank, is also needed for filling them. One of the feed pumps, of which there should always be at least two, as already stated, is fitted with a tee in the suction pipe near the pump to receive the pipe

leading from the tanks. During the test the main suction leading to this pump from the general supply should be kept closed, so that only the water that passes through the tanks is used for feeding the boiler. If the plant be a small one, with but one or two boilers and only a single feed pump, the latter can be made to do duty as a testing pump, because during the test there will be no other boilers to feed besides the ones under test.

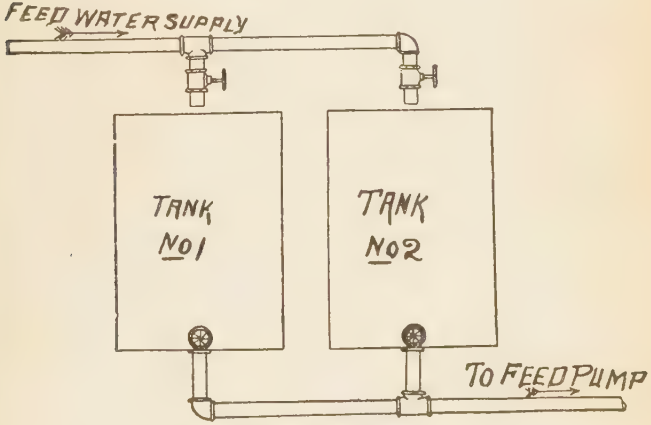


FIGURE 10.

If metal tanks are considered too expensive, two good water-tight barrels can be substituted. Fig. 10 will give the reader a general idea of what is needed for obtaining the weight of the water by the method just described. If a closed heater is used and no other boilers are in service during the test, the cold water can be measured in the tanks and pumped directly through the heater, but if it is necessary to feed other boilers besides those under test, then either a separate

feed pipe must be run to the test boilers, or else hot water meters will have to be put into the branch pipes.

In cases where a separate feed pipe must be put in for the test boiler and the water which is used for testing cannot be passed through a heater, there should be a ¾ or 1 in. pipe connected to the feed main or header and leading to the testing tanks, in order to allow a portion of the hot feed water to run into and mix with the cold water in the tanks as they are being filled, thus partially warming the water before it goes to the boiler.

Heating Surface. The heating surface of a boiler consists of that portion of the boiler which is exposed to the heat on one side and water on the other. In a horizontal boiler of either the flue or tubular type, the available heating surface is, first, the lower half of the shell; second, the area of the back head below the water line minus the combined cross sectional area of all the tubes or flues; third, the inside area of the flues; fourth, the area of the front head minus the sectional area of the flues.

For a fire-box boiler of the vertical type, the area of the flue sheets minus the sectional area of the flues, plus the area of the fire-box plus the inside area of the flues constitutes the heating surface. If the boiler is a horizontal internally fired boiler, the heating surface will consist of, first, area of three sides of the fire-box; second, area of the crown sheet; third, area of flue sheets minus sectional area of flues; fourth, inside area of the flues.

In estimating the area of the fire-box, the area of the fire door should be subtracted therefrom. If the fire-box be circular, as in the case of a vertical boiler, the area may be obtained by first finding by measurements

the diameter, which multiplied by 3.1416 will give the circumference. Then multiply this result by the height or the distance between the grate bars and the flue sheet. In the case of water tube boilers the outside area of the tubes must be taken. Two examples will be given illustrating methods of calculating heating surface:

First, take a horizontal tubular boiler, diameter 72 in., length 18 ft., having sixty-two 4½ in. flues; find area of lower half of shell.

Circumference = diameter × 3.1416 = 18.8496 ft.

One-half of the circumference multiplied by the length = required area. Thus, 18.8496 ÷ 2 × 18 = 169.64 sq. ft.

Next find heating surface of back head below the water line. Total area = $72^2 \times .7854$ = 4071.5 sq. in. Assume two-thirds of this area to be exposed to the heat. ⅔ of 4071.5 = 2714.3 sq. in. From this must be deducted the sectional area of the tubes. In giving the size of boiler tubes the outside diameter is taken. The tubes being 4½ in.; the area of a circle 4½ in. in diameter is 15.9 sq. in. Number of flues, 62 × 15.9 = 985.8 sq. in. = sectional area of tubes. The heating surface of the back head therefore = 2714.3 − 985.8 = 1728.5 sq. in. Dividing this by 144, to reduce to feet, we have 12 sq. ft.

Next find inside area of tubes. The standard thickness of a 4½ in. tube = .134 in. The inside diameter therefore will be 4.5 − (2 × .134) = 4.23 in., and the circumference will be 4.23 × 3.1416 = 13.29 in., and the inside area will be 13.29 × length, 18 ft., = 216 in. Thus 216 × 13.29 ÷ 144 = 19.93 sq. ft., inside area of one flue. There being 62 flues, the total heating surface of tubes is 19.93 × 62 = 1235.66 sq. ft. The heating sur-

face of the front head is found in the same manner as that of the back head, with the exception that the whole area should be figured instead of two-thirds, for the reason that the entire surface is exposed to the heat, although that portion above the water line may be considered as superheating surface. The heating surface of front head would be: area 4071.5 − sectional area of tubes 785.8 = 3285.7 sq. in. = 21.43 sq. ft.

The total heating surface of the boiler is thus found to be 1438.73 sq. ft, divided up as follows:

Lower half of shell,	169.64	sq. ft.
Back head,	12.00	"
Tubes,	1235.66	"
Front head,	21.43	"
	1438.73	"

Next taking a vertical fire-box boiler of the following dimensions: diameter of flue sheet, and also of fire-box, 50 in.; height of fire-box above grate bars, 30 in.; number of flues, 200; size of flues, 2 in.; length of flues, 7 ft.

First, find heating surface in flue sheet.

Area of circle, 50 in. in diameter = 1,963.5 sq. in.

Sectional area of 2 in. flue = 3.14 sq. in., which multiplied by 200 = 628 sq. in., total sectional area of tubes. The heating surface of one flue sheet therefore will be $1{,}963.5 - 628 \div 144 = 9$ sq. ft.

Assuming that the tops of the flues are submerged, the area of the top flue sheet will also be 9 sq. ft. Then heating surface of flue sheets = $9 \times 2 = 18$ sq. ft.

Second, find heating surface of tubes. The standard thickness of a 2 in. flue is .095 in. The inside diameter will consequently be $2 - (.095 \times 2) = 1.8$ in., and the circumference will be $1.8 \times 3.1416 = 5.66$ in. The

length of the flue being 7 ft., or 84 in., the inside area will be $5.66 \times 84 \div 144 = 3.3$ sq. ft., and multiplying this result by 200 we have $200 \times 3.3 = 660$ sq. ft. as the heating surface of the flues.

Third, find heating surface of the fire-box. Diameter of fire-box = 50 in., which multiplied by $3.1416 = 157.08$, which is the circumference. The height being 30 in., the total area will be $157.08 \times 30 \div 144 = 32.7$ sq. ft. Allowing 1 sq. ft. as the area of the fire door, will leave 31.7 sq. ft. heating surface of fire-box. The heating surface of the boiler will be:

For the flue sheets,	18	sq. ft.
For the flues,	660	"
For the fire-box,	31.7	"
Total,	709.7	"

The above methods may be applied in estimating the heating surface of any boiler, provided in the case of water tube boilers that the outside in place of the inside area of the tubes be figured.

Questions

1. How should the bridge wall of a horizontal boiler be built?

2. How should the brick work of a boiler be secured in order to prevent damage by expansion and contraction?

3. Which end of a horizontal boiler should be the lowest, and why?

4. How should the water column be located?

5. How high above the top row of tubes should the lower gauge cock be?

6. What is the proper ratio of safety valve area to grate surface?

7. Where should the fusible plug be located?

8. Where should the feed pipe enter the boiler?

9. Where should the blow off pipe be connected?

10. What is the most economical device for feeding a boiler?

11. In selecting a feed pump, how may the required size of pump be ascertained?

12. What is the disadvantage in the use of the injector for feeding a boiler?

13. What is gained by using a feed water heater?

14. How many kinds of exhaust heaters are there?

15. How may the saving effected by using the exhaust steam for heating the feed water be estimated?

16. What should the capacity of the heater be?

17. What provision should be made for testing coal and other fuel?

18. What is the heating surface of a boiler?

Answers

1. Straight across from wall to wall, and from 8 to 10 inches below bottom of boiler.

2. By leaving an air space of 2 inches in the rear and side walls, beginning at, or near, the level of the grate bars and extending up to the fire-line or about the center line of boiler. Above this height the walls should be solid.

3. The back end should be at least one inch lower than the front end in order that the rear ends of the tubes may always be covered with water so long as any appears in the gauge glass.

4. The lower end of the gauge glass should be level with the top of the upper row of tubes. This refers to horizontal tubular boilers.

5. About 3 inches.

6. For lever and weight valves; 1 sq. in. of valve area for each 2 sq. ft. of grate surface. For pop valves; 1 sq. in. of valve area for each 3 sq. ft. of grate surface.

7. In that part of the heating surface which is first liable to be overheated from lack of water.

8. At that point where the feed water is least liable to come directly in contact with the hot boiler sheets.

9. With the lowest portion of the water space.

10. The belt-driven power pump is the most economical but not the safest or most convenient.

11. Ascertain number of square feet of heating surface in the boiler, or boilers. Then allow a pump capacity of one cu. ft. of water per hour for each 15 sq. ft. of heating surface.

12. An injector will not work well with the feed water at a temperature much higher than 100 degrees F.

13. The utilization of all available exhaust steam for heating the feed water before it enters the boiler.

14. Two kinds; open, and closed.

15. First, find temperature of feed water before it enters heater. Second, find its temperature as it leaves the heater. Third, ascertain by the steam table the number of heat units above 32 degrees F. in the water at each of the two temperatures. Fourth, subtract the less from the greater. The remainder will be the number of heat units added to the water by means of the heater.

16. The capacity to supply the boiler for 15 or 20 minutes. A body of water would then be continually in contact with the heating surface and, passing slowly through, it will receive much more heat than if rushed through too small a heater.

17. A tee should be located in the feed pipe as near the check valve as practicable and between it and the feed pump. One opening of this tee is reduced to ¼ or ½ inch to receive a hot water thermometer for showing at what temperature the feed water enters the boiler. Means should also be provided for ascertaining the weight of water fed to the boiler during the test, and for weighing the coal or other fuel consumed.

18. All that portion of the boiler exposed to heat on one side and to water on the other.

CHAPTER V

BOILER OPERATION

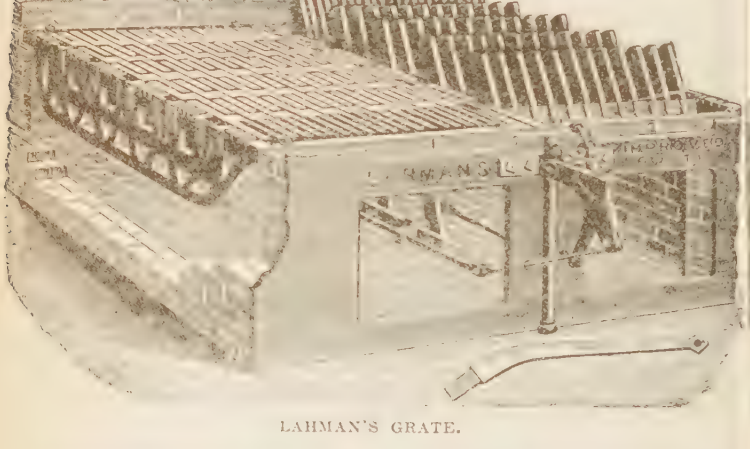
First care of the engineer on entering his boiler-room—Cleaning fires—Fire tools, etc.—Firing—Suggestions as to best method of firing—Quantity of air required per pound of coal—Cleaning tubes—Washing out, etc.—Why it is dangerous to cool a boiler too quickly—Repairing tubes—Cleaning inside of boiler—Pitting—How to feed a boiler—What to do in cases of emergency—Connecting with main—Foaming, priming, etc.—Safety valve calculations—Rules for safety valve calculations—Feed pumps—Care of feed pumps—Directions for setting steam valves of duplex pumps—Hydraulics for engineers.

Operation. Having considered in the previous chapters the principal details in the construction and erection of boilers with which the working engineer is interested, it is now in order to devote a space to their operation.

Duties. The first act of the careful engineer on entering his boiler-room when he goes on duty should be to ascertain the exact height of the water in his boilers. This he can do by opening the valve in the drain pipe of the water column, allowing it to blow out freely for a few seconds, then close it tight and allow the water to settle back in the glass. This should be done with each boiler under steam, not only once, but several times during the day. No engineer should be satisfied with a general squint along the line of gauge glasses, but he should either go himself or else instruct his fireman or water tender to make the rounds of each boiler and be sure that the water is all right.

The next thing to be looked after is the fire. If the plant is run continuously day and night it is the duty of the firemen coming off watch to have the fires clean, the ash pits all cleaned out, a good supply of coal on the floor, and everything in good order for the on coming force. A good fireman will take pride in always leaving things in neat shape for the man who is to relieve him.

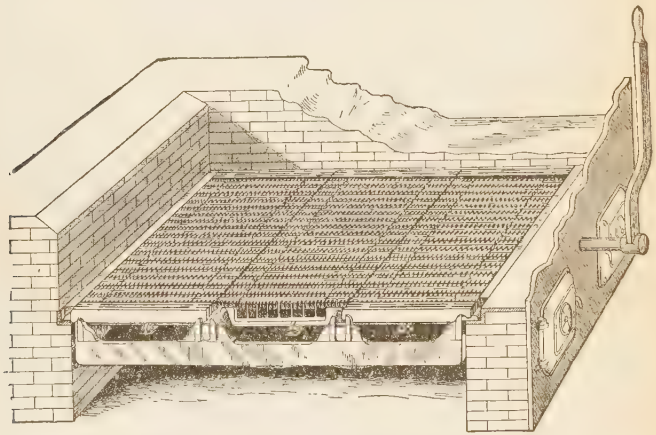
Cleaning Fires. With some varieties of coal this is



LAHMAN'S GRATE.

a comparatively easy task, especially if the boilers are fitted with shaking grates. With a coal that does not form a clinker on the grate bars, the fires can be kept in good condition by cleaning them twice or three times in twenty-four hours, as the larger part of the loose ashes and noncombustible can be gotten rid of by shaking the grates and using the slice bar at intervals more or less frequent; but such coals are generally considered too expensive to use in the ordinary manufacturing plant, and cheaper grades are substituted.

Fire Tools. For cleaning fires successfully and quickly the following tools should be provided: a slice bar, a fire hook, a heavy iron or steel hoe, and a light hoe for cleaning the ash-pit. It is unnecessary to describe these tools, as they are familiar to all engineers. A suggestion as to the kind of handles with which they should be fitted may be of benefit. The working ends of the aforesaid tools having been made and each welded to a bar of 1 or 1⅛ in. round



MARTIN ANTI-FRICTION ROCKING GRATES.

iron and 10 or 12 in. long, take pieces of 1 or 1¼ in. iron pipe cut to the length desired for the handles and weld the shanks of the tools to them. To the other end of the pipe weld a handle made of round iron somewhat smaller than the shank. By using pipe handles the weight of the tools is considerably lessened, and they will still be sufficiently strong. The labor of cleaning the fire will thus be greatly lightened. When a fire shows signs of being foul and choked with

clinker, preparations should be made at once for cleaning it by allowing one side to burn down as low as possible, putting fresh coal on the other side alone. When the first side has burned as low as it can without danger of letting the steam pressure fall too much, take the slice bar and run it in along the side of the furnace on top of the clinker and back to near the bridge wall, then using the door jamb as a fulcrum, give it a quick strong sweep across the fire and the greater part of the live coals will be pushed over to the other side. What remains of the coal not yet consumed can be pulled out upon the floor with the light hoe and shoveled to one side, to be thrown back into the furnace after the clinker is taken out. Having now disposed of the live coal, take the slice bar and run it along on top of the grates, loosening and breaking up the clinker thoroughly, after which take the heavy hoe and pull it all out on the floor. A helper should be ready with a pail of water, or, what is still better, a small rubber hose connected to a cold water pipe running along the boiler fronts for this purpose, and put on just enough water to quench the intense heat of the red hot clinker as it lies on the floor. When the grates are cleaned, close the door, and with the slice bar in the other side push all the live coal over to the side just cleaned, where it should be leveled off and fresh coal added. After this has become ignited, treat the other side in the same way. An expert fireman will thus clean a fire with very little loss in steam pressure, and practically no waste of coal.

Firing. No definite set of rules for hand firing can be laid down that will be suitable for all steam plants, or for the many different kinds of coal used. Some kinds of coal need very little stirring or slicing, while

others that have a tendency to coke and form a crust on top of the fire need to be sliced quite often.

Every engineer, if he is at all observant, should be able to judge for himself as to the best method of treating the coal he is using, so as to get the most economical results. A few general maxims may be laid down. First, keep a clean fire; second, see that every square inch of grate surface is covered with a good live fire; third, keep a level fire, don't allow hills and valleys and yawning chasms to form in the furnace, but keep the fire level; fourth, when cleaning the fire always be sure to clean all the clinkers and dead ashes away from the back end of the grates at the bridge wall, in order that the air may have a free passage through the grate bars, because this is one of the best points in the furnace for securing good combustion provided the bridge wall is kept clean from the grates up. By keeping the back ends of the grate bars and the face of the bridge wall clean, the air is permitted to come in contact with the hot fire brick, and thus one of the greatest aids to good combustion is utilized. Don't allow the fire to become so deep and heavy that the air cannot pass up through it, because without a good supply of air good combustion is impossible. When the chimney draft is good the quantity of cold air admitted underneath the grate bars may be easily regulated by leaving the ash-pit doors partly open. The amount of opening required can be ascertained by a little experimenting and depends upon the intensity of the draft and the condition of the fire. With a clean, light fire and the air spaces in the grates free from dead ashes, a slight opening of the ash-pit doors will suffice to admit all the air required beneath the grates. But if the fire is

heavy and the grates are clogged, a larger opening will be necessary. In firing bituminous coal containing a large percentage of volatile (light or gaseous matter) the best results can be obtained by leaving the fire doors slightly open for a few seconds immediately after throwing in a fresh fire. The reason for doing this is that the volatile matter in the coal flashes into flame the instant it comes in contact with the heat of the furnace, and if a sufficient supply of oxygen is not present just at this particular time the combustion will be imperfect and the result will be the formation of carbon mon-oxide or carbonic oxide gas, and the loss of about two-thirds of the heat units contained in the coal. This loss can be guarded against in a great measure by a sufficient volume of air, either through the fire doors directly after putting in a fresh fire, or, what is still better, providing air ducts through the bridge wall or side walls which will bring the air in on top of the fire. Each pound of coal requires for its complete combustion 12 lbs. or about 150 cu. ft. of air, and the largest volume of air is needed just after fresh coal has been added to the fire.

Cleanliness. In order to get the best results great care should be taken that the tubes be kept clean and free from soot. Especially does this apply to horizontal return tubular boilers, for the reason that when the tubes become clogged with soot the efficiency of the draft is destroyed and the steaming capacity of the boiler is greatly reduced. Soot not only stops the draft, but it is a non-conductor of heat. In some batteries of boilers where an inferior grade of coal is used and the draft is poor, it is absolutely necessary to scrape or blow the tubes at least once a day in order to enable the boilers to generate sufficient steam.

As to the process of cleaning there are various devices on the market, both for blowing the soot out by means of a steam jet and also for scraping the inside of the tubes. The steam jet, if properly made and used with a high pressure and dry steam, does very satisfactory work, but it should not be depended upon exclusively to keep the tubes clean, because in process of time a scale will form inside the tubes that nothing but a good scraper will remove. For that reason it is good practice to use the scraper two or three times a week at least. When the boiler is cooled down for washing out, the bottom of the shell should be cleaned of all accumulations of dust and ashes, the combustion chamber back of the bridge wall cleaned out, and the back flue sheet or head swept off and examined, and if there is a fusible plug in the back head the scale should be scraped from it, both inside and outside the boiler, because if it is covered with scale neither the water nor the heat can come in contact with it, and it will be non-effective.

Washing Out. The length of time that a boiler can be run safely and economically after having been washed out depends upon the nature of the feed water. If the water is impregnated to a considerable extent with scale forming matter, the boiler should be washed out every two weeks at the least, and in some cases of particularly bad water it becomes necessary to shorten the time to one week. To prepare a boiler for washing the fire should be allowed to burn as low as possible and then be pulled out of the furnace, the furnace doors left slightly ajar and the damper left wide open in order that the walls may gradually cool. It is as bad a practice to cool a boiler off too suddenly as it is to fire it up too quick, because the sudden change of

temperature either way has an injurious effect on the seams, contracting or expanding the plates, according as it is cooled or warmed, and thus creating leaks and very often small cracks radiating from the rivet holes, and becoming larger with each change of temperature, until finally the strength of the seam is destroyed and rupture takes place. After the boiler has become comparatively cool and there is no pressure indicated by the steam gauge the blow off cock may be opened and the water allowed to run out. The gauge cocks and also the drip to the water column should be left open to allow the air to enter and displace the water. Otherwise there will be a partial vacuum formed in the boiler and the water will not run out freely.

A boiler should not be blown out, that is, emptied of water while under pressure. The sudden change of temperature is sure to have a bad effect upon the sheets and seams. Suppose for instance that all the water is blown out of a boiler under a pressure of 20 lbs. by the steam gauge. The temperature of steam at 20 lbs. is 200° F., and it may be assumed that the metal of the boiler is at or near that temperature also. Assume the temperature of the atmosphere in the boiler-room to be 60° F. There will then be a range of 260° (60° – 200°) temperature for the boiler to pass through within a short time, which will certainly have a bad effect, and besides this the boiler shell will be so hot that the loose mud and sediment left after the water has run out is liable to be baked upon the sheets, making it much harder to remove.

While inside the boiler the boiler washer should closely examine all the braces and stays, and if any are found loose or broken they should be repaired at once before the boiler is used again. The soundness of

braces, rivets, etc., can be ascertained by tapping them with a light hammer.

Renewing Tubes. As it is practically impossible to prevent scale from forming on the outside of the tubes of horizontal tubular boilers unless the feed water is exceptionally good, and as the tubes will in course of time become leaky where they are expanded into the heads, the engineer if he has a battery of two or more, should take advantage of the first opportunity that presents itself to take out of service the boiler that shows the most signs of deterioration and take out the tubes, and after cleaning them of scale by scraping and hammering or rolling in a tumbling cylinder, he should select those that are still in good condition and have them pieced out at the ends, making them almost as good as new.

The flues being out of the boiler will give the boiler washer a good opportunity to thoroughly clean the inside also, and if there are any loose rivets they should be replaced and leaky or suspicious looking seams chipped and caulked. If there are indications of corrosion or pitting, a stiff paste or putty made of plumbago mixed with a small proportion of cylinder oil may be applied to the affected parts with good results.

Feed Water. There is no steam plant of any consequence that does not have more or less exhaust steam or returns from a steam heating system which can be utilized for heating the feed water before it enters the boiler. Cold water should never be pumped into a boiler that is under steam when it is possible to prevent it.

In feeding a boiler the speed of the feed pump should be so gauged as to supply the water just as fast

as it is evaporated. The firing can then be even and regular.

If the supply of feed water should suddenly be cut off, owing to breakage of the pump or bursting of a water main, and no other source of supply was available, the dampers should be immediately closed, or if there should be no damper in the breeching, the draft may be stopped by opening the flue doors. The fires should then be deadened by shoveling wet or damp ashes in on top of them, or if the ashes cannot be readily procured, bank the fires over with green coal broken into fine bits. This, with the draft all shut off, will deaden the fires, while the engine still running will gradually use up the extra steam. If the water should get dangerously low in the boilers the fires may be pulled, provided they have become deadened sufficiently, but they should never be pulled while they are burning lively, because the stirring will only serve to increase the heat and the danger will be aggravated.

Connecting a Recently Fired Up Boiler. After a boiler has been washed out, filled with water, and fired up, the next move is to connect it with the main battery. The steam in the boiler to be connected having been raised to the same pressure as that in the battery, the connecting valve should be opened slightly, just enough to permit a small jet of steam to pass through, which can be heard by placing the ear near the body of the valve. This jet of steam may be passing from the battery into the newly connected boiler or *vice versa*. Whichever way it passes, the valve should not be opened any farther until the flow of steam stops, which will indicate that the pressure has been equalized. It will then be found that the valve will move

much easier and it may be gradually opened until it is wide open.

Foaming. Water carried with the steam from the boiler to the engine, even if in small quantities, is very detrimental to the successful operation of the engine, as it washes the oil from the walls of the cylinder, thereby increasing the friction, and unless a plentiful supply of oil is entering the cylinder cutting of the piston rings will take place. There is also danger of breaking a cylinder head or of bending the piston rod if the water comes in too large quantities.

There are certain kinds of water which have a natural tendency to foam, especially such as contain considerable organic matter, and the more severe the service to which the boiler is put the more will the water foam, until it is practically impossible to locate the true level of the water in the boilers, and the only recourse the water tender has is to keep his feed pump running at such a speed as will in his judgment supply the water as fast as it goes out of the boilers. It is a dangerous condition to say the least, and the only remedy for it is either a change to a different kind of water, or if this is not possible, then an increase in the number of boilers, which would make it possible to supply sufficient steam for the engine without being compelled to fire the boilers so hard.

Priming. By which is meant the carrying over of water in the form of fine spray mingled with the steam, is not so dangerous as foaming and yet it causes much loss in the efficiency of a boiler or engine. It can be prevented to a large extent by placing a baffle plate in the steam space of the boiler directly under the dome or outlet to the connection with the steam main.

Safety Valves. Rules are given in Chapter IV. for guidance in making calculations relating to spring pop valves, which are now almost universally used on boilers, and which, without doubt, are the most reliable appliance for relieving a boiler of surplus steam.

A short space will be devoted to the consideration of the lever safety valve also, as it may be of interest to some students.

The U. S. marine rule for lever valves is here repeated: "Lever safety valves to be attached to marine boilers shall have an area of not less than one square inch to every two square feet of grate surface in the boiler, and the seats of all such safety valves shall have an angle of inclination of 45° to the center line of their axis."

In order to arrive at accurate results in lever safety valve calculations it is necessary to know first the number of pounds pressure exerted upon the stem of the valve by the lever itself, irrespective of the weight, also the weight of the valve and stem, as all these weights together with the weight of the ball suspended upon the lever tend to hold the valve down against the pressure of the steam. The effective weight of the lever can be ascertained by leaving it in its position attached to the fulcrum and connecting a spring balance scale to it at the point where it rests on the valve stem. The weight of the valve and stem can also be found by means of the scale. When the above weights are known, together with the weight at the end of the lever and its distance from the fulcrum, also the area of the valve and its distance from the fulcrum, the pressure at which the valve will blow can be found by the following rules:

Rule 1. Multiply the weight by its distance from the

fulcrum. Multiply the weight of the valve and lever by the distance of the stem from the fulcrum and add this to the former product. Divide the sum of the two products by the product of the area of the valve multiplied by the distance of its stem from the fulcrum. The result will be pressure in pounds per square inch required to lift the valve.

Example. Diameter of value, 3 in.

Distance of stem from fulcrum, 3 in.

Effective weight of lever, valve and stem, 20 lbs.

Weight of ball, 50 lbs.

Distance of ball from fulcrum, 30 in.

Required pressure at which the valve will blow off, $50 \times 30 + 20 \times 3 = 1560$.

Area of valve, $7.0686 \times 3 = 21.2058$.

$1560 \div 21.2058 = 73.57$ lbs. pressure.

When the pressure at which it is desired the valve should blow off is known, together with the weights of all the parts the proper distance from the fulcrum at which to place the weight is ascertained by Rule 2.

Rule 2. Multiply the area of the valve by the pressure and from the product subtract the effective weight of the valve and lever. Multiply the remainder by the distance of the stem from the fulcrum and divide by the weight of the ball. The quotient will be the required distance.

Example. Area of valve, 7.07 sq. in.

To blow off at 75 lbs.

Effective weight of lever and valve, 20 lbs.

Weight of ball, 50 lbs.

Distance of valve stem from fulcrum, 3 in.

$7.07 \times 75 - 20 = 510.25$.

$510.25 \times 3 \div 50 = 30.6$ in., distance from fulcrum at which to place the ball.

When the pressure is known, together with the distance of the weight from the fulcrum, the weight of the ball is obtained by Rule 3.

Rule 3. Multiply the area of the valve by the pressure and from the product subtract the effective weight of the lever and valve. Multiply the remainder by the distance of the stem from the fulcrum and divide by the distance of the ball from the fulcrum. The quotient will be the required weight.

Example. Area of valve, ... 7.07 sq. in.
Pressure in pounds per square inch, . . 80 lbs.
Effective weight of lever and valve, . . 20 lbs.
Distance of stem from fulcrum, ... 3 in.
Distance of weight from fulcrum, . . . 30 in.

$7.07 \times 80 - 20 = 545.6$.

$545.6 \cdot 3 \div 30 = 54.56$ lbs., weight of ball.

Safety valves, especially those of the lever type, are liable to become corroded and stick to their seats if allowed to go any great length of time without blowing. Therefore it is good practice to raise the steam pressure to the blowing off point at least two or three times a week, or oftener, for the purpose of testing the valve. If it opens and releases the steam at the proper point all is well, but if it does not, it should be looked after forthwith. Generally the mere raising of the lever by hand, or a few taps with a hammer if it be a pop valve, will free it and cause it to work all right again; but if this treatment has to be resorted to very often the valve should be taken down and overhauled. In too many steam plants not enough importance is attached to the safety valve. The fact is, it is one of the most useful and important adjuncts of a boiler, and if neglected serious results are sure to follow.

Feed Pumps. A good engineer will always take a pride in keeping his feed pump in good condition, and if he has two or more of them, which every steam plant of any consequence should have, he will have an opportunity to keep his pumps in good shape. The water pistons of most boiler feed pumps are fitted to receive rings of fibrous packing. The best packing for this purpose and one that will stand both hot and cold water service is made of pure canvas cut in strips of the required width, ½, ⅝, ¾ in., etc., and laid together with a water proof cement having the edges for the wearing surface. This packing is called square canvas packing, and can be purchased in any size required for the pump. The size is easily ascertained by placing the water piston, minus the follower plate, centrally in the water cylinder and measuring the space between the piston and cylinder walls. This packing should not be allowed to run for too long a time before renewing, for the reason that pieces of it are liable to become loose and be forced along with the feed water on its way to the boiler and lodge under the check valve, holding it open and causing no end of trouble. If the feed pump has to handle hot water, or has to lift the water several feet by suction, the packing rings should be looked after at least once a month.

Hard rubber valves are, all things considered, the best for a boiler feed pump, as they are not affected by hot water and do not hammer the seats like metallic composition valves do. Every boiler feed pump should be fitted with a good sight-feed lubricator for cylinder oil. The steam valve mechanism of a steam pump is very sensitive and delicate and requires good lubrication in order to do good work. In too many

cases reed pumps are fitted with an old style cylinder oil cup and there is generally more oil on the outside of the valve chest than there is inside, while the valve is bulldozed into working by frequent blows from a convenient club.

The steam valves of all steam pumps are adjusted before they are sent out from the factory, and most of them are arranged so that the stroke may be shortened or lengthened as the engineer desires. It is best as a rule to allow a pump to make as long a stroke as it will without striking the heads, because then the parts are worn evenly.

Sometimes an engineer is called upon to set the valves of a duplex pump which have become disarranged. In such a case he should proceed as follows: Place both pistons exactly at mid-stroke. This may be done in two ways. First, by dropping a plummet line alongside the levers connecting the rock shafts with the spools on the piston rods. Then bring the rods to the position where the centers of the spools will be in a vertical line with the centers of the rock shafts.

The second method is to move the piston to the extreme end of the stroke until it comes in contact with the cylinder head. Then mark the rod at the face of the stuffing box gland. Next move the piston to the other end of the stroke and mark the rod at the opposite gland. Now make a mark on the rod exactly half way between the two outside marks and move the piston back until the middle mark is at the face of the gland and the piston will be at mid-stroke. Having placed both pistons at mid-strike, remove the valve chest covers and adjust the valves in their central position, viz., so that they cover the steam ports. The

valve rod being in position and connected to the rocker arm by means of the short link, the nut or nuts securing the valve to the rod should be so adjusted as to be equidistant from the lugs on the valve, say $\frac{3}{32}$ or $\frac{1}{8}$ of an inch according to the amount of lost motion desired, which latter factor governs the length of stroke in some makes of duplex pumps, while in others it is controlled by tappets on the valve rod outside of the valve chest. Care should be taken while making these adjustments that the valve be retained exactly in its central position.

Having set the valves correctly, move one of the pistons far enough from mid-stroke to get a small opening of the steam port on the opposite side, then replace the valve chest covers and the pump will be ready to run. As these valves are generally made without any outside lap, a slight movement of one of the pistons in either direction from its central position will suffice to uncover one of the ports on the other cylinder sufficiently to start the pump.

Sometimes duplex pumps "work lame," that is, one piston will make a quick full stroke while the other piston will move very slowly and just far enough to work the steam valve of the opposite side. In the majority of cases this irregular action is due to unequal friction in the packing of the rods, or the packing rings on one of the pistons may be worn out.

If one side of a duplex pump becomes disabled from any cause, as breaking of piston rod in the water cylinder, for instance, which is liable to happen, the pump may still be operated in the following manner until duplicate parts to replace the broken ones have been secured. Loosen the nuts or tappets on the valve stem of the broken side and place them far

enough apart so that the steam valve will be moved through only a small portion of its stroke, thereby admitting only steam enough to move the empty steam piston and rod, and thus work the steam valve of the remaining side. The packing on the broken rod should be screwed up tight, so as to create as much friction as possible; there being no resistance in the water end. In this way the pump may be operated for several days or weeks and thus prevent a shut down.

Hydraulics for Engineers. Among the many difficult problems that are continually coming up for engineers to solve, there is none more perplexing than the correct calculation of the quantity of water which will be discharged in a given time from pipes of various sizes and under the many different heads or pressures. Problems in hydraulics, as given by the majority of writers on engineering, are usually in elaborate algebraical equations, which, to the ordinary working engineer, are very perplexing, at least the author has found them to be so in his experience. Therefore with a view of assisting his brother engineers in the solution of problems along this line which they may be called upon to solve, the author has spent considerable time and labor in searching for and compiling a few rules and examples for hydraulic calculations in plain arithmetic which he hopes may be of benefit.

First, to find velocity of flow in the pump, or in other words, piston speed.

Rule. Multiply number of strokes per minute by length of stroke in feet, or fractions thereof.

Second, the velocity of flow in the discharge pipe is in inverse ratio to the squares of the diameters of the pipe and the water cylinder of pump.

Thus, a pump cylinder is 6 in. in diameter, and the piston speed is 100 ft. per minute; the discharge pipe being 3 in. in diameter. What is the velocity of flow in the pipe?

Example. $\frac{6\times6}{3\times3} = 4$. In this case the velocity in the pipe is four times that in the pump, and $100 \times 4 = 400$ ft. per minute, velocity for water in the discharge pipe.

Third, to find velocity in feet per minute necessary to discharge a given quantity of water in a given time.

Rule. Multiply the number of cubic feet to be discharged by 144 and divide by area of pipe in inches.

Fourth, to find area of pipe when the volume and velocity of water to be discharged are known.

Rule. Multiply volume in cubic feet by 144 and divide by the velocity in feet per minute.

Fifth, one of the first requisites in making correct calculations of the quantity of water discharged from any sized pipe is to obtain the velocity of flow per second. There are several rules for doing this, among which the following appear to be the plainest and most simple:

Rule 1. Multiply the square root of the head in inches by the constant 27.8. For instance, assume the head to be 100 ft. = 1200 in. The square root of 1200 is 35 nearly, then $35 \times 27.8 = 973$ in. = 81 ft. per second velocity.

Rule 2. Multiply the square root of the head in feet by the constant 8, as follows: The square of 100 = 10 and $10 \times 8 = 80$ ft. velocity per second.

Rule 3. Multiply twice the acceleration of gravity by the head in feet and extract the square root of product. The acceleration of gravity may be considered the constant number 32, neglecting decimals.

$32 \times 2 \times 100 = 6400$. Square root of $6400 = 80$ ft. per second.

In many instances it is more convenient to use the pressure in pounds per square inch as shown by gauge instead of the height or head, and we can then apply Rule 4.

Rule 4. Multiply the square root of the pressure in pounds per square inch by the constant number 12.16 as follows: Pressure due to 100 ft. head = 44 lbs., nearly. Square root of 44 = 6.6, which multiplied by $12.16 = 80.2$ ft. velocity per second.

Having ascertained the velocity of flow, we may now proceed to calculate the weight of water in pounds per second discharged from any size of pipe, neglecting for the time being the loss in pressure caused by friction from elbows and bends in the pipe and also the peculiar shape assumed by a stream of water flowing through pipes or conduits when there is no resistance except the pressure of the atmosphere and friction caused by long distance transmission.

We will take for our calculation a four-inch pipe from which the water has a free flow under a head of 100 ft., which gives a velocity of 80 ft. per second.

Rule 5. Divide the velocity in feet per second by the constant 2.3, and multiply the quotient by the area of discharge pipe in square inches. $80 \div 2.3 = 34.7$. Now the area of a four-inch pipe is 12.57 sq. in., and $34.7 \times 12.57 = 436$ lbs. discharged per second.

In order to get the matter clearly before us, let us assume that we have a section of four-inch pipe just 80 ft. in length and that it lies in a horizontal position and is filled solidly full of water. It will contain area, 12.57 sq. in. × length, 960 in. = 12,067.2 cu. in. of water, and as one pound of water occupies a space of

27.7 cu. in., we therefore have 1,2067.2 ÷ 27.7 = 436 lbs. of water, and at a velocity of 80 ft. per second our pipe will be emptied and refilled continuously each second. We have also Rule 6 to find the number of cubic feet discharged per minute when the velocity per minute is known.

Rule 6. Multiply the area of pipe in square inches by the velocity in feet per minute and divide by the constant 144.

Example. Area of 4 in. pipe = 12.57 sq. in. Velocity of flow = 80 ft. per second = 4,800 ft. per minute. Then $\frac{12.57 \times 4.800}{144}$ = 419 cu. ft. per minute = 6.99 cu. ft. per second, which multiplied by 62.3 lbs. (weight of 1 cu. ft.) = 435.4 lbs. per second.

As stated before, no allowance is made by the above rules for friction or other retarding influences, but for ordinary purposes in connection with a steam plant a deduction of 25 per cent. is probably sufficient. Of course if the water is being discharged into an elevated tank or against direct pressure of any kind, the resistance in pounds per square inch or the height in feet must be deducted from the impelling pressure or head. Let us assume, for instance, that our 4 in. pipe is discharging water into a tank at an elevation of 75 ft. above the level of the pump, and that to reach the tank requires 100 ft. of pipe with two 90° ells and one straight-way valve. We wish to discharge 500 gal. per minute into the tank and will therefore require a velocity of about 13 ft. per second, which will necessitate a pressure of a little more than 1 lb. per square inch to be maintained at the pump over and above all resistance. Now the resistance to be overcome in this case will be:

Pressure per square inch due to 75 ft. head,	32.5 lbs.
Friction loss due to length of pipe and velocity,	7.43 "
Friction loss due to two 90° ells,	2.16 "
Friction loss due to straight way valve,	.2 "
Total,	42.29 lbs.

Requiring a pressure of say 43 lbs. per square inch, or about the equivalent of 100 ft. head at the pump.

Again, suppose that in place of the elevated tank we have 1,000 ft. of 8 in. horizontal pipe with a 4 in. delivery at the end farthest from the pump, and three branch pipes each 100 ft. long and 4 in. in diameter with one 90° ell and one straight-way valve, connected at intervals to the 8 in. main, and it is required to discharge in all 1,000 gals. per minute, or at the rate of 250 gals. per minute for each 4 in. delivery. The friction loss for each 100 ft. in length of 8 in. pipe at a velocity of 13 ft. per second is .94 lbs., and for each 100 ft. of 4 in. pipe it is 1.89 lbs. Likewise the friction loss for each 90° ell is 1.08 lbs., and for a straight way valve .2 lbs., at the above velocity. The total resistance therefore to be overcome is as follows:

For 1,000 ft. of 8 in. pipe,	.94 lbs. × 10	= 9.4 lbs.
For 300 ft. of 4 in. pipe,	1.89 lbs. × 3	= 5.67 "
For four 90° ells,	1.08 lbs. × 4	= 4.32 "
For four straight-way valves,	.2 lbs. × 4	= .8 "
Total,		20.19 lbs.

Consequently the pressure required at the pump will be about 22 lbs. per square inch, equal to a head of 50 ft.

Questions

1. What should be the first care of an engineer on entering his boiler room when he goes on watch?

2. What is one of the duties of the fireman before coming off watch?

3. Name the various tools necessary for cleaning fires properly.

4. Describe the operation of cleaning fires.

5. What general rules should be observed in firing?

6. How may the best results be obtained in firing bituminous coal?

7. What will be the result if a sufficient supply of oxygen is not admitted to the furnace?

8. What quantity of air (cubic feet or pounds weight) is required for the complete combustion of one pound of coal?

9. Suppose there is a fusible plug in the boiler and it becomes covered with scale, what will be the result?

10. How long may a boiler be run safely after washing it out?

11. What should be done with a boiler in order to prepare it for washing out?

12. What effect does the too sudden cooling off or firing up of a boiler have upon the seams?

13. What other bad result takes place within the boiler when it is emptied of water while hot?

14. What should be done with tubes that have become badly scaled?

15. How fast should the feed water be supplied?

16. What should be done in case the supply of feed water be cut off suddenly?

17. What is the proper method of connecting a recently fired up boiler to the main header?

18. What are some of the dangerous results of foaming?

19. What can be done to prevent or at least to modify foaming in boilers?

20. Which are the most reliable pop valves or lever safety valves?

21. What is the rule regulating the area of lever safety valves?

22. What two factors must first be known before correct calculations can be made as to the weight and position of the ball on a lever safety valve?

23. How may the effective weight of the lever and that of the valve and stem be found?

24. When the area in square inches of the valve, the weight of the valve and stem, the effective weight of the lever, the weight of the ball and its distance from the fulcrum are known, how is the pressure at which the valve will blow off ascertained?

25. When the pressure at which the valve should blow off is known, together with the weights of all the parts, how may the distance the ball should be from the fulcrum be ascertained?

26. When the pressure is known, together with the distance of the ball from the fulcrum, how is the weight of the ball found?

27. What should be done with a safety valve in order to keep it in good working condition?

28. Describe the process of setting the steam valves of a duplex pump.

29. How may a duplex pump be operated in case one of the water pistons becomes disabled?

30. How is the velocity of flow or piston speed per minute of a pump ascertained?

31. The piston speed being known, how is the velocity of flow in the discharge pipe found?

32. When it is required to discharge a certain quantity of water from a given size of pipe in a given time, how may the velocity of flow in feet per minute be found?

33. When the volume of water to be discharged and the velocity of flow are known, how is the area of the pipe obtained?

34. What is meant by "acceleration of gravity," and what constant number represents it in connection with hydraulics?

35. What per cent. of allowance is ordinarily made for friction in water pipes?

Answers

1. To ascertain the exact height of the water in the boilers. This can be done by opening the valve in drain pipe of each water column and allowing it to blow freely for a few seconds, then closing it tight and allowing the water to settle back in the gauge glass.

2. To leave clean fires and ash pits, a good supply of coal on the floor and everything in good order for the man who is to relieve him.

3. A slice bar, a fire hook, a heavy iron or steel hoe, and a light hoe for cleaning the ash pit.

4. First, allow one side of the fire to burn down as low as possible without danger of letting the steam pressure drop too low; then run the slice bar in alongside the furnace wall, on top of the clinker and back nearly to the bridge wall and with the door jamb as a fulcrum give the bar a quick, strong sweep across the fire, thus pushing the greater part of the live coals over to the other side. Second, having now disposed of the live coal, take the slice bar and run it along on top of the grates, loosening and breaking up the clinker, after which take the heavy hoe and pull it all out on the floor. Third, close the fire door and again using the slice bar push all the live coal over to the side just cleaned.

Then level it off and add fresh coal. When this becomes ignited, clean the other side.

5. First, keep a clean fire. Second, see that every square inch of grate surface is covered with a good live fire. Third, keep a level fire, but don't allow the fire to become so deep and heavy that the air cannot pass up through it. Fourth, when cleaning the fire always be sure to clean all the clinkers and dead ashes away from the back end of the grates at the bridge wall. Fifth, with a good chimney draft the proper supply of cold air to be admitted underneath the grate bars can be regulated by leaving the ash pit doors partly open.

6. By leaving the fire doors slightly open for a few seconds immediately after throwing in a fresh fire.

7. Imperfect combustion, resulting in the formation of carbon monoxide, and the loss of about two-thirds of the heat units contained in the coal.

8. Twelve pounds, or about 150 cubic feet.

9. Neither water nor heat can come in contact with it, and it will be useless as a safety appliance.

10. Not longer than two weeks.

11. Allow the fire to burn as low as possible, then pull it out of the furnace, after which leave the furnace doors slightly open and the damper wide open in order that the walls may gradually cool.

12. An injurious effect. It contracts or expands the plates, according to whether it is cooled or warmed, thus creating leaks and small cracks radiating from the rivet holes.

13. The loose mud and sediment remaining in the boiler after the water has run out is liable to be baked upon the sheets, making it much harder to remove.

14. They should be taken out, cleaned of all scale, and those that are in good condition may be pieced out at the ends, making them almost as good as new.

15. Just as fast as it is evaporated.

16. Close all dampers and shut off all draft. Then deaden the fires by shoveling in wet or damp ashes on top of them, or if ashes are not readily available, bank the fires over with green coal broken into fine bits.

17. The steam pressure in the boiler to be connected should equal that in the main header. The connecting valve should then be opened slightly, just enough to permit a small jet of steam to pass through, which may be heard by placing the ear near the valve body. The valve should then be left in this position until the flow of steam stops. This will indicate that the pressure has become equalized, and the valve may now be gradually opened until it is wide open.

18. Water carried over into the engine cylinder washes the oil from the cylinder walls and the piston rings are liable to cut. There is also danger of breaking a cylinder head or of bending a piston rod if the water comes in large quantities.

19. Either change to a different kind of feed water or else increase the number of boilers in service.

20. Pop valves.

21. They should have an area of not less than one sq. in. to each two sq. feet of grate surface.

22. The number of pounds pressure exerted upon the valve stem by the lever itself without the weight of the ball; also the weight of the valve and stem.

23. Place the lever in its position attached to the fulcrum and connect a spring balance scale to it at the point where it rests upon the valve stem. The weight of the

valve and stem can also be found by means of the scale.

24. Multiply weight of ball by its distance from fulcrum. Multiply weight of valve and lever by distance of stem from fulcrum and add this to former product. Divide the sum of the two products by the product of area of valve multiplied by distance of stem from fulcrum. The result will be pressure in lbs. per sq. inch at which the valve will blow off.

25. Multiply area of valve by pressure and from this product subtract weight of valve and lever. Multiply remainder by distance of stem from fulcrum and divide product by weight of ball. Result will be distance of ball from fulcrum.

26. Multiply area of valve by pressure and from the product subtract weight of lever and valve. Multiply remainder by distance of stem from fulcrum and divide product by distance of ball from fulcrum. Result will be weight of ball.

27. It should be tested two or three times each week by raising the steam pressure to the blowing off point. If the valve does not open promptly it should be attended to at once.

28. First, place both pistons at mid-stroke. This can be done by dropping a plummet line alongside the levers connecting the rock-shafts with the spools on the piston rods. Then bring the rods to the position where the centers of the spools will be in a vertical line with the centers of the rock-shafts. Second, with both pistons at mid-stroke remove the valve chest covers and adjust the valves in their central position, viz., so that they cover the steam ports. Third, adjust the nuts securing the valve to the rod so that they will be equidistant from the lugs on the valve, this distance to be 3/32 or 1/8 of an inch. Fourth,

having thus set the valves, move one piston far enough from mid-stroke to get a small opening of the steam port on the opposite side, then replace the valve chest covers and the pump will be ready to run.

29. Loosen the nuts or tappets on the valve stem of broken side and adjust them far enough apart to allow a movement of the valve through only a small part of its stroke, thereby admitting only sufficient steam to move the empty piston and rod and thus work the steam valve on the good side of the pump. The piston rod packing on the broken side should be screwed up tight so as to create as much friction as possible, there being no resistance in the water end.

30. Multiply number of strokes per minute by length of stroke in feet or fractions thereof.

31. Square diameter in inches of water cylinder of pump; also square diameter in inches of the discharge pipe. Then divide the first result by the second and multiply by piston speed of pump in feet per minute. The result will be velocity of flow in feet per minute in discharge pipe.

32. Multiply number of cu. feet to be discharged per minute by 144 and divide by area of pipe in square inches.

33. Multiply volume in cu. feet to be discharged per minute by 144 and divide by velocity of flow in feet per minute.

34. In the case of falling bodies the acceleration due to gravity is 32.16 feet per second. This constant is used in calculations of pressures and velocity of water due to head or height in feet above sea level.

35. For ordinary purposes in connection with the operation of a steam plant, a deduction of 20 to 25 per cent is sufficient.

CHAPTER VI

COMBUSTION—WATER—STEAM

Combustion—Composition of air—Carbon—The principal constituent of fuels—Hydrogen, its nature and heating value—Table, giving analysis of various coals—Process of combustion described—Quantity of air required—Furnace temperature—How to utilize the heat in the escaping gases—Heat, what it is and how produced—Joule's researches. The heat unit—Specific heat—Sensible heat and latent heat—Experiments of Professor Black—Total heat of evaporation—Water, its composition, nature, etc.—Steam, its expansive nature, temperature, etc.—Saturated steam, etc.—Total heat of steam—Density of steam.

Combustion. The subject of the combustion of fuels being one in which every engineer is vitally interested, it is proper that its leading features and principles be discussed. One of the main factors in the combustion of coal is the proper supply of air. Air is composed of two gases, oxygen and nitrogen, in the proportion, by volume, of 21 per cent. of oxygen and 79 per cent. of nitrogen, or by weight, 23 per cent. of oxygen and 77 per cent. of nitrogen.

The composition of pure dry air, as given by Kent in *Steam Boiler Economy*, is as follows:

By volume, 20.91 parts O. and 79.09 parts N.

By weight, 23.15 parts O. and 76.85 parts N.

Air is a mixture and not a chemical combination of these two elements. The principal constituent of coal and most other fuels, whether solid, liquid or gaseous, is carbon. Hydrogen is a light combustible gas and combined either with carbon or with carbon and

oxygen, in various proportions, is also a valuable constituent of fuels, notably of bituminous coal. The heating value of one pound of pure carbon is rated at 14,500 heat units, while one pound of hydrogen gas contains 62,000 heat units.

Analysis of coal shows that it contains moisture, fixed carbon, volatile matter, ash and sulphur in various proportions according to the quality of the coal. The following table deduced from a few of the many valuable tables of analysis of the coals of the United States, as given by Mr. Kent, will show the composition of the principal bituminous coals in use in this country for steam purposes. Two samples are selected from each of the great coal producing states, with the exception of Illinois, from which four were taken.

TABLE 2

State	Kind of Coal	Moisture	Volatile Matter	Fixed Carbon	Ash	Sulphur
Pennsylvania	Youghiogheny	1.03	36.49	59.05	2.61	0.81
"	Connellsville	1.26	30.10	59.61	8.23	0.78
West Virginia	Quinimont	0.76	18.65	79.26	1.11	0.23
"	Fire Creek	0.61	22.34	75.02	1.47	0.56
E. Kentucky	Peach Orchard	4.60	35.70	53.28	6.42	1.08
"	Pike County	1.80	26.80	67.60	3.80	0.97
Alabama	Cahaba	1.66	33.28	63.04	2.02	0.53
"	Pratt Co.'s	1.47	32.29	59.50	6.73	1.22
Ohio	Hocking Valley	6.59	35.77	49.64	8.00	1.59
"	Muskingum "	3.47	37.88	53.30	5.35	2.24
Indiana	Block	8.50	31.00	57.50	3.00	
"	"	2.50	44.75	51.25	1.50	
W. Kentucky	Nolin River	4.70	33.24	54.04	11.70	2.54
"	Ohio County	3.70	30.70	45.00	3.16	1.24
Illinois	Big Muddy	6.40	30.60	54.60	8.30	1.50
"	Wilmington	15.50	32.80	39.90	11.80	
"	" screenings	14.00	28.00	34.20	23.80	
"	Duquoin	8.90	23.50	60.60	7.00	

The process of combustion consists in the union of the carbon and hydrogen of the fuel with the oxygen

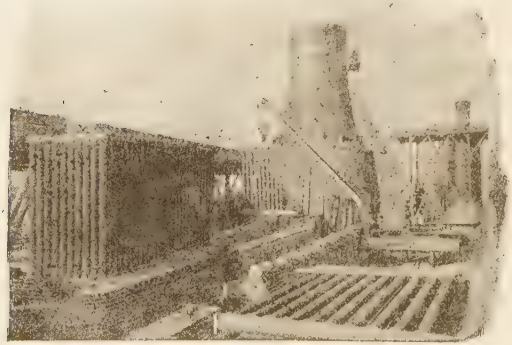
of the air. Each atom of carbon combines with two atoms of oxygen, and the energetic vibration set up by their combination is heat. Bituminous coal contains a large percentage of volatile matter which is released and flashes into flame when the coal is thrown into the furnace, and unless air is supplied in large amounts at this stage of the combustion there will be an excess of smoke and consequent loss of carbon. On the other hand there is a loss in admitting too much air because the surplus is heated to the temperature of the furnace without aiding the combustion and will carry off to the chimney just as many heat units as were required to raise it from the temperature at which it entered the furnace to that at which it enters the uptake. It will therefore be seen that a great advantage will be gained by first allowing the air that is needed above the fire to pass over or through heated bridge walls or side walls. Some kinds of coal need more air for their combustion than do others, and good judgment and close observation are needed on the part of the fireman to properly regulate the supply.

Some boilers will make steam more economically by partly closing the ash pit doors, while others require the same doors to be kept wide open. The quantity of air required for the combustion of one pound of coal is, by volume, about 150 cu. ft.; by weight, about 12 lbs.

The temperature of the furnace is usually about 2500°, in some cases reaching as high as 3000°. The temperature of the escaping gases should not be much above nor below 400° F. for bituminous coal. The waste heat in the escaping gases can be utilized to great advantage by passing them through what are called economizers before they escape into the chim-

ney. These economizers consist of coils or stacks of cast iron pipe placed within the flue or breeching leading from the boilers to the chimney and are enveloped in the hot gases, while the feed water is passed through the pipes on its way to the boilers, the result being that considerable heat is thus imparted to the feed water that would otherwise go to waste.

In order to attain the highest economy in the burning of coal in boiler furnaces two factors are indispen-



GREEN'S FUEL ECONOMIZER UNDER CONSTRUCTION.

sable, viz., a constant high furnace temperature and quick combustion, and these factors can only be secured by supplying the fresh coal constantly just as fast as it is burned, and also by preventing as much as possible the admission of cold air to the furnace. This is why the automatic or mechanical stoker, if it be of the proper design, is more economical and causes less smoke than hand firing. The fireman when he puts in a fire is prone to shovel in a good supply all at once, and this has the tendency to

greatly reduce the temperature of the furnace, while at the same time it retards combustion. On the other hand the mechanical stoker supplies the coal continuously only as fast as it is required and no faster, and the furnace doors do not need to be opened at all, by which a large volume of cold air is prevented from entering the furnace and reducing the temperature. The author does not wish to be understood as recommending the adoption and use of mechanical stokers to replace hand firing, but he draws this contrast between the two methods of firing in order that it may be of some benefit to the thousands of honest toilers who earn a livelihood by shoveling coal into boiler furnaces.

The problem of the economical use of coal and the abatement of the smoke nuisance, especially in our large cities, has of late years become so serious that it is to the interest of every engineer, and especially every fireman, to use the utmost diligence, care and good judgment in the use of coal, and to emulate as much as possible the methods of the mechanical stoker.

Heat. All matter, whether solid, liquid or gaseous, consists of molecules or atoms, which are in a state of continual vibration, and the result of this vibration is heat. The intensity of the heat evolved depends upon the degree of agitation to which the molecules are subject.

Until as late as the beginning of the nineteenth century two rival theories in regard to the nature of heat had been advocated by scientists. The older of these theories was that heat was a material substance, a subtle elastic fluid termed caloric, and that this fluid penetrated matter something like water penetrates a

sponge. But this theory was shown to be false by the wonderful researches and experiments of Count Rumford at Munich, Bavaria, in 1798.

By means of the friction between two heavy metallic bodies placed in a wooden trough filled with water, one of the pieces of metal being rotated by machinery driven by horses, Count Rumford succeeded in raising the temperature of the water in two and one-half hours from its original temperature of 60° to 212° F., the boiling point, thus demonstrating that heat is not a material substance, but that it is due to vibration or motion, an internal commotion among the molecules of matter. This theory, known as the Kinetic theory of heat, has since been generally accepted, although it was nearly fifty years after Rumford advocated it in a paper read before the Royal Society of Great Britain in 1798, before scientists generally became converted to this idea of the nature of heat and the science of Thermo Dynamics placed on a firm basis.

During the period from 1840 to 1849 Dr. Joule made a series of experiments which not only confirmed the truth of Count Rumford's theory that heat was not a material substance but a form of energy which may be applied to or taken away from bodies, but Joule's experiments also established a method of estimating in mechanical units or foot pounds the amount of that energy. This latter was a most important discovery because by means of it the exact relation between heat and work can be accurately measured.

The first law of thermo dynamics is this: Heat and mechanical energy or work are mutually convertible. That is, a certain amount of work will produce a certain amount of heat, and the heat thus produced is capable of producing by its disappearance a fixed

amount of mechanical energy if rightly applied. The mechanical energy in the form of heat which, through the medium of the steam engine, has revolutionized the world, was first stored up by the sun's heat millions of years ago in the coal which in turn, by

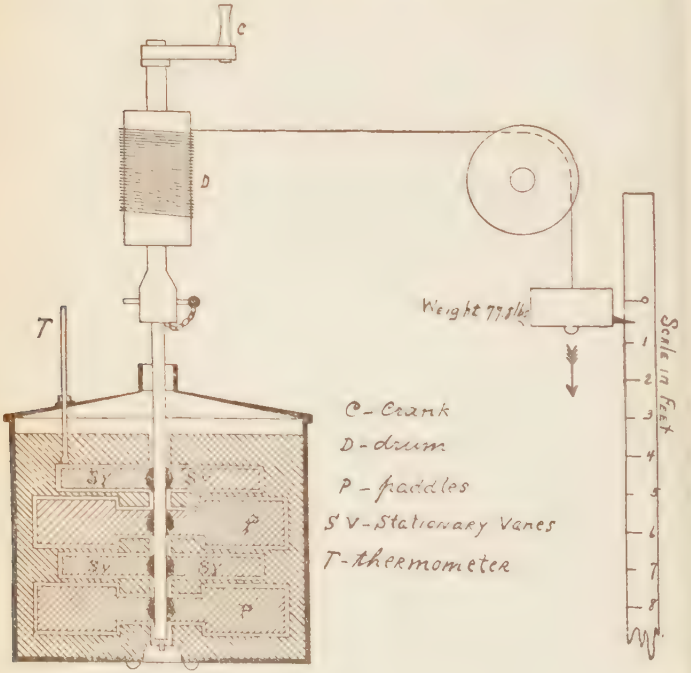


FIGURE 11

combustion, is made to release it for purposes of mechanical work.

The general principles of Dr. Joule's device for measuring the amount of work in heat are illustrated in Fig. 11. It consisted of a small copper cylinder

containing a known quantity of water at a known temperature. Inside the cylinder and extending through the top was a vertical shaft to which were fixed paddles for stirring the water. Stationary vanes were also placed inside the cylinder. Motion was imparted to the shaft through the medium of a cord or small rope coiled around a drum near the top of the shaft and running over a grooved pulley or sheave. To the free end of the cord a known weight was attached. This weight was allowed to fall through a certain distance, and in falling it turned the shaft with its paddles, which in turn agitated the water, thus producing a certain amount of heat. To illustrate, suppose the weight to be 77.8 lbs., and that by means of the crank at the top end of the shaft it has been raised to the zero mark at the top of the scale. (See Fig. 11.) One pound of water at 39.1° F. is poured into the copper cylinder, which is then closed and the weight released. At the moment the weight passes the 10 ft. mark on the scale, the thermometer attached to the cylinder will indicate that the temperature of the water has been raised one degree. Then multiplying the number of pounds in the weight by the distance in feet through which it fell will give the number of foot pounds of work done. Thus, 77.8 lbs. × 10 ft. = 778 foot pounds.

The heat unit or British thermal unit (B. T. U.) is the quantity of heat required to raise the temperature of one pound of water one degree, or from 39° to 40° F., and the amount of mechanical work required to produce a unit of heat is 778 foot pounds. Therefore the mechanical equivalent of heat is the energy required to raise 778 lbs. one foot high, or 77.8 lbs. 10 ft. high, or 1 lb. 778 feet high. Or again, suppose

a one-pound weight falls through a space of 778 ft. or a weight of 778 lbs. falls one foot, enough mechanical energy would thus be developed to raise a pound of water one degree in temperature, provided all the energy so developed could be utilized in churning or stirring the water, as in Joule's machine. Hence the mechanical equivalent of heat is 778 foot pounds.

Specific Heat. The specific heat of any substance is the ratio of the quantity of heat required to raise a given weight of that substance one degree in temperature to the quantity of heat required to raise an equal weight of water one degree in temperature when the water is at its maximum density, 39.1° F. To illustrate, take the specific heat of lead, for instance, which is .031, while the specific heat of water is 1. That means that it would require 31 times as much heat to raise one pound of water one degree in temperature as it would to raise the temperature of a pound of lead one degree.

The following table gives the specific heat of different substances in which engineers are most generally interested.

TABLE 3.

Water at 39.1° F.	1.000
Ice at 32° F.	.504
Steam at 212° F.	.480
Mercury	.033
Cast iron	.130
Wrought iron	.113
Soft steel	.116
Copper	.095
Lead	.031
Coal	.240
Air	.238
Hydrogen	3.404
Oxygen	.218
Nitrogen	.244

Sensible Heat and Latent Heat. The plainest and most simple definition of these two terms is that given

by Sir Wm. Thomson. He says: "Heat given to a body and warming it is sensible heat. Heat given to a body and not warming it is latent heat." Sensible heat in a substance is the heat that can be measured in degrees of a thermometer, while latent heat is the heat in any substance that is not shown by the thermometer.

To illustrate this more fully a brief reference to some experiments made by Professor Black in 1762 will no doubt make the matter plain. It will be remembered that at that early date comparatively little was known of the true nature of heat, hence Professor Black's investigations and discoveries along this line appear all the more wonderful. He procured equal weights of ice at 32° F. and water at the same temperature, that is, just at the freezing point, and placing them in separate glass vessels suspended the vessels in a room in which the uniform temperature was 47° F. He noticed that in one-half hour the water had increased 7° F. in temperature, but that twenty half hours elapsed before all of the ice was melted. Therefore he reasoned that twenty times more heat had entered the ice than had entered the water, because at the end of the twenty half hours when the ice was all melted the water in both vessels was of the same temperature. The water having absorbed 7° of heat during the first half hour must have continued to absorb heat at the same rate during the whole of the twenty half hours, although the thermometer did not indicate it. From this he calculated that $7° \times 20 = 140°$ of heat had become latent or hidden in the water.

In another experiment Professor Black placed a lump of melting ice, which he estimated to be at a

temperature of 33° F. on the surface, in a vessel containing the same weight of water at 176° F., and he observed that when the whole of the ice had been melted the temperature of the water was 33° F., thus proving that 143° of heat (176 − 33) had been absorbed in melting the ice and was at that moment latent in the water. By these two experiments Professor Black established the theory of the latent heat of water, and his estimate was very near the truth because the results obtained since that time by the greatest experimenters show that the latent heat of water is 142 heat units, or B. T. U.

Black's experiment for ascertaining the latent heat in steam at atmospheric pressure was made in the following simple manner: He placed a flat, open tin dish on a hot plate over a fire and into the dish he put a small quantity of water at 50° F. In four minutes the water began to boil, and in twenty minutes more it had all evaporated. In the first four minutes the temperature had increased 212° − 50° = 162°, and the temperature remained at 212° throughout the twenty minutes that it required to evaporate all the water, despite the fact that the water had been receiving heat during this period at the same rate as during the first four minutes. He therefore reasoned that in the twenty minutes the water had absorbed five times as much heat as it had in the four minutes, or 162° × 5 = 810°, without any sensible rise in temperature. Therefore the 810° became latent in the steam. Owing to the crude nature of the experiment Professor Black's estimate of the number of degrees of latent heat in steam was incorrect, as it has been proven by many famous experimenters since then that the latent heat of steam at atmospheric pressure is 965.7 B. T. U.

It will thus be perceived that what is meant by the term latent heat is that quantity of heat which becomes hidden or latent when the state of a body is changed from a solid to a liquid, as in the case of melting ice, or from a liquid to a gaseous state, as with water evaporated into steam. But the heat so disappearing has not been lost, on the contrary it has, while becoming latent, been doing an immense amount of work, as can easily be ascertained by means of a few simple figures. It has been seen that a heat unit is the quantity of heat required to raise one pound of water one degree in temperature and also that the mechanical equivalent of heat, or, in other words, the mechanical energy stored in one heat unit is equal to 778 foot pounds of work.

A horse power equals 33,000 ft. lbs. of energy in one minute of time, and a heat unit = 778 ÷ 33,000 = .0236, or about $\frac{1}{43}$ of a horse power. The work done by the heat which becomes latent in converting one pound of ice at 32° F. into water at the same temperature = 142 heat units × 778 ft. lbs. = 110,476 ft. lbs., which divided by 33,000 equals 3.34 horse power. Again, by the evaporation of one pound of water from 32° F. into steam at atmospheric pressure, 965.7 units of heat become latent in the steam and the work done = 965.7 × 778 = 751,314 ft. lbs. = 22.7 horse power. It will thus be seen what tremendous energy lies stored in one pound of coal, which contains from 12,000 to 14,500 heat units, provided all the heat could be utilized in an engine.

Total Heat of Evaporation. In order to raise the temperature of one pound of water from the freezing point, 32° F., to the boiling point, 212° F., there must be added to the temperature of the water 212° − 32° =

180°. This represents the sensible heat. Then to make the water boil at atmospheric pressure, or, in other words, to evaporate it, there must still be added 965.7 B. T. U., thus 180 + 965.7 = 1,145.7, or in round numbers 1,146 heat units. This represents what is termed the total heat of evaporation at atmospheric pressure and is the sum of the sensible and latent heat in steam at that pressure. But if a thermometer were held in steam evaporating into the open air, as, for instance, in front of the spout of a tea-kettle, it would indicate but 212° F.

When steam is generated at a higher pressure than 212°, the sensible heat increases and the latent heat decreases slowly, while at the same time the total heat of evaporation slowly increases as the pressure increases, but not in the same ratio. As, for instance, the total heat in steam at atmospheric pressure is 1,146 B. T. U., while the total heat in steam at 100 lbs. gauge pressure is 1,185 B. T. U., and the sensible temperature of steam at atmospheric pressure is 212°, while at 100 lbs. gauge pressure the temperature is 338 and the latent heat is 876 B. T. U. See table.

Water. The elements that enter into the composition of pure water are the two gases, hydrogen and oxygen, in the following proportions:

By volume, hydrogen 2, oxygen 1.
By weight, " 11.1, " 88.9.

Perfectly pure water is not attainable, neither is it desirable nor necessary to the welfare of the human race, because the presence of certain proportions of air and ammonia add greatly to its value as an agent for manufacturing purposes and for generating steam. The nearest approach to pure water is rain water, but even this contains 2.5 volumes of air to each 100 vol-

umes of water. Pure distilled water, such for instance as the return water from steam heating systems, is not desirable for use alone in a boiler as it will cause corrosion and pitting of the sheets, but if it is mixed with other water before going into the boiler its use is highly beneficial, as it will prevent to a certain degree the formation of scale and incrustation. Nearly all water used for the generation of steam in boilers contains more or less scale-forming matter, such as the carbonates of lime and magnesia, the sulphates of lime and magnesia, oxide of iron, silica and organic matter, which latter tends to cause foaming in boilers.

The carbonates of lime and magnesia are the chief causes of incrustation. The sulphate of lime forms a hard crystalline scale which is extremely difficult to remove when once formed on the sheets and tubes of boilers. Of late years the intelligent application of chemistry to the analyzing of feed waters has been of great benefit to engineers and steam users, in that it has enabled them to properly treat the water with solvents either before it is pumped into the boiler, or by the introduction into the boiler of certain scale preventing compounds made especially for treating the particular kind of water used. Where it is necessary to treat water in this manner great care and watchfulness should be exercised by the engineer in the selection and use of a boiler compound.

From ten to forty grains of mineral matter per gallon are held in solution by the waters of the different rivers, streams and lakes; well and mine water contain still more.

Water contracts and becomes denser in cooling until it reaches a temperature of 39.1° F., its point of greatest density. Below this temperature it expands and

at 32° F. it becomes solid or freezes, and in the act of freezing it expands considerably, as every engineer who has had to deal with frozen water pipes can testify.

Water is 815 times heavier than atmospheric air. The weight of a cubic foot of water at 39.1° is approximately 62.5 lbs., although authorities differ on this matter, some of them placing it at 62.379 lbs., and others at 62.425 lbs. per cubic foot. As its temperature increases its weight per cubic foot decreases until at 212° F. one cubic foot weighs 59.76 lbs.

The table which follows is compiled from various sources and gives the weight of a cubic foot of water at different temperatures.

TABLE 4

Temperature	Weight per Cubic Foot	Temperature	Weight per Cubic Foot	Temperature	Weight per Cubic Foot
32° F.	62.42 lbs.	132° F.	61.52 lbs.	230° F.	59.37 lbs.
42°	62.42	142°	61.34	240°	59.10
52°	62.40	152°	61.14	250°	58.85
62°	62.36	162°	60.94	260°	58.52
72°	62.30	172°	60.73	270°	58.21
82°	62.21	182°	60.50	300°	57.26
92°	62.11	192°	60.27	330°	56.24
102°	62.00	202°	60.02	360°	55.16
112°	61.86	212°	59.76	390°	54.03
122°	61.70	220°	59.64	420°	52.86

The boiling point of water varies according to the pressure to which it is subject. In the open air at sea level the boiling point is 212° F. When confined in a boiler under steam pressure the boiling point of water depends upon the pressure and temperature of the steam, as, for instance, at 100 lbs. gauge pressure the temperature of the steam is 338° F., to which temper-

ature the water must be raised before its molecules will separate and be converted into steam. In the absence of any pressure, as in a perfect vacuum, water boils at 32° F. temperature. In a vacuum of 28 in., corresponding to an absolute pressure of .943 lbs., water will boil at 100°, and in a vacuum of 26 in., at which the absolute pressure is 2 lbs., the boiling point of water is 127° F. On the tops of high mountains in a rarefied atmosphere water will boil at a much lower temperature than at sea level, for instance at an altitude of 15,000 ft. above sea level water boils at 184° F.

Steam. Having discussed to some extent the physical properties of water, it is now in order to devote some time to the study of the nature of steam, which is simply water in its gaseous form made so by the application of heat.

As has been stated in another portion of this book, matter consists of molecules or atoms inconceivably small in size, yet each having an individuality, and in the case of solids or liquids, each having a mutual cohesion or attraction for the other, and all being in a state of continual vibration more or less violent according to the temperature of the body.

The law of gravitation which holds the universe together, also exerts its wonderful influence on these atoms and causes them to hold together with more or less tenacity according to the nature of the substance. Thus it is much more difficult to chip off pieces of iron or granite than it is of wood. But in the case of water and other liquids the atoms, while they adhere to each other to a certain extent, still they are not so hard to separate, in fact, tney are to some extent repulsive to each other, and unless confined within certain bounds

the atoms will gradually scatter and spread out, and finally either be evaporated or sink out of sight in the earth's surface. Heat applied to any substance tends to accelerate the vibrations of the molecules, and if enough heat is applied it will reduce the hardest substances to a liquid or gaseous state.

The process of the generation of steam from water is simply an increase of the natural vibrations of the molecules of the water, caused by the application of heat until they lose all attraction for each other and become instead entirely repulsive, and unless confined will fly off into space. But being confined they continually strike against the sides of the containing vessel, thus causing the pressure which steam or any other gas exerts when under confinement.

Of course steam, like other gases, when under pressure is invisible, but the laws governing its action are well known. These laws, especially those relating to the expansion of steam, will be more fully discussed in the chapters on the Indicator. The temperature of steam in contact with the water from which it is generated, as for instance in the ordinary steam boiler, depends upon the pressure under which it is generated. Thus at atmospheric pressure its temperature is 212° F. If the vessel is closed and the pressure increased the temperature of the steam and also that of the water rises.

Saturated Steam. When steam is taken directly from the boiler to the engine without being superheated, it is termed saturated steam. This does not necessarily imply that it is wet and mixed with spray and moisture.

Superheated Steam. When steam is conducted into or through a vessel or coils of pipe separate from the

boiler in which it was generated and is there heated to a higher temperature than that due to its pressure, it is said to be superheated.

Dry Steam. When steam contains no moisture it is said to be dry. Dry steam may be either saturated or superheated.

Wet Steam. When steam contains mist or spray intermingled it is termed wet steam, although it may have the same temperature as dry saturated steam of the same pressure.

During the further consideration of steam in this book, saturated steam will be mainly under discussion for the reason that this is the normal condition of steam as used most generally in steam engines.

Total Heat of Steam. The total heat in steam includes the heat required to raise the temperature of the water from 32° F. to the temperature of the steam plus the heat required to evaporate the water at that temperature. This latter heat becomes latent in the steam, and is therefore called the latent heat of steam.

The work done by the heat acting within the mass of water and causing the molecules to rise to the surface is termed by scientists internal work, and the work done in compressing the steam already formed in the boiler or in pushing it against the superincumbent atmosphere, if the vessel be open, is termed external work. There are, therefore, in reality three elements to be taken into consideration in estimating the total heat of steam, but as the heat expended in doing external work is done within the mass itself it may, for practical purposes, be included in the general term latent heat of steam.

Density of Steam. The expression density of steam means the actual weight in pounds or fractions of a

pound avoirdupois of a given volume of steam, as one cubic foot. This is a very important point for young engineers especially to remember, so as not to get the two terms, pounds pressure and pounds weight, mixed, as some are prone to do.

Volume of Steam. By this term is meant the volume as expressed by the number of cubic feet in one pound weight of steam.

Relative Volume of Steam. This expression has reference to the number of volumes of steam produced from one volume of water. Thus the steam produced by the evaporation of one cubic foot of water from 39° F. into steam at atmospheric pressure will occupy a space of 1640 cu. ft., but, as the steam is compressed and the pressure allowed to rise, the relative volume of the steam becomes smaller, as for instance at 100 lbs. gauge pressure the steam produced from one cubic foot of water will occupy but 237.6 cu. ft., and if the same steam was compressed to 1,000 lbs. absolute or 985.3 lbs. gauge pressure it would then occupy only 30 cu. ft.

The condition of steam as regards its dryness may be approximately estimated by observing its appearance as it issues from a pet cock or other small opening into the atmosphere. Dry or nearly dry steam containing about 1 per cent. of moisture will be transparent close to the orifice through which it issues, and even if it is of a grayish white color it may be estimated to contain not over 2 per cent. of moisture.

Steam in its relation to the engine should be considered in the character of a vehicle for transferring the energy, created by the heat, from the boiler to the engine. For this reason all steam drums, headers and pipes should be thoroughly insulated in order to

prevent, as much as possible, the loss of heat or energy by radiation.

Table 5, showing the properties of saturated steam, is compiled mainly from Kent's *Steam Boiler Economy*. The decimals have not been carried out in the columns headed Relative Volume and Weight of 1 cu. ft. of steam, as their absence will affect the results very little.

TABLE 5

PROPERTIES OF SATURATED STEAM

Vacuum Inches of Mercury	Absolute Pressure Lbs. per Sq. Inch	Temp. Degrees F.	Total Heat above 32° F.		Latent Heat H-h Heat units	Relative Volume	Cubic Feet in 1 Lb. Wt. of Steam	Wt. of 1 Cubic Foot of Steam, Lbs.
			In the Water h Heat-units	In the Steam H Heat-units				
29.74	.089	32.	0.	1091.7	1091.7	208,080	3333.3	.0003
29.67	.122	40.	8.	1094.1	1086.1	154,330	2472.2	.0004
29.56	.176	50.	18.	1097.2	1079.2	107,630	1724.1	.0006
29.40	.254	60.	28.01	1100.2	1072.2	76,370	1223.4	.0008
29.19	.359	70.	38.02	1103.3	1065.3	54,660	875.61	.0011
28.90	.502	80.	48.04	1106.3	1058.3	39,690	635.80	.0016
28.51	.692	90.	58.06	1109.4	1051.3	29,290	469.20	.0021
28.00	.943	100.	68.08	1112.4	1044.4	21,830	349.70	.0028
27.88	1.	102.1	70.09	1113.1	1043.0	20,623	334.23	.0030
25.85	2.	126.3	94.44	1120.5	1026.0	10,730	173.23	.0058
23.83	3.	141.6	109.9	1125.1	1015.3	7,325	118.00	.0085
21.78	4.	153.1	121.4	1128.6	1007.2	5,588	89.80	.0111
19.74	5.	162.3	130.7	1131.4	1000.7	4,530	72.50	.0137
17.70	6.	170.1	138.6	1133.8	995.2	3,816	61.10	.0163
15.67	7.	176.9	145.4	1135.9	990.5	3,302	53.00	.0189
13.63	8.	182.9	151.5	1137.7	986.2	2,912	46.60	.0214
11.60	9.	188.3	156.9	1139.4	982.4	2,607	41.82	.0239
9.56	10.	193.2	161.9	1140.9	979.0	2,361	37.80	.0264
7.52	11.	197.8	166.5	1142.3	975.8	2,159	34.61	.0289
5.49	12.	202.0	170.7	1143.5	972.8	1,990	31.90	.0314
3.45	13.	205.9	174.7	1144.7	970.0	1,846	29.60	.0338
1.41	14.	209.6	178.4	1145.9	967.4	1,721	27.50	.0363
0.00	14.7	212.0	180.9	1146.6	965.7	1,646	26.36	.0379

TABLE 5—*Continued*

Gauge Pressure Lbs. per Sq. In.	Absolute Pressure Lbs. per Sq. In.	Temp. Degrees F.	Total Heat Above 32° F.		Latent Heat H Heat-units	Relative Volume	Cubic Feet in 1 Lb. Wt. of Steam	Wt. of 1 Cubic Foot of Steam, Lbs.
			In the Water h Heat-units	In the Steam H Heat-units				
0.3	15	213.3	181.9	1146.9	[illegible]	1,014	25.90	.0387
1.3	16	216.3	185.3	1147.9	[illegible]	1,519	24.33	.0411
2.3	17	219.4	188.4	1148.9	[illegible]	1,434	23.00	.0435
3.3	18	222.4	191.4	1149.8	[illegible]	1,354	21.80	.0459
4.3	19	225.2	194.3	1150.6	[illegible]	1,292	20.70	.0483
5.3	20	227.9	197.0	1151.5	[illegible]	1,243	19.72	.0507
6.3	21	230.5	199.7	1152.2	[illegible]	1,197	18.84	.0531
7.3	22	233.0	202.2	1153.0	[illegible]	1,156	18.03	.0555
8.3	23	235.4	204.7	1153.7	[illegible]	1,118	17.30	.0578
9.3	24	237.8	207.0	1154.5	[illegible]	1,083	16.62	.0602
10.3	25	240.0	209.3	1155.1	[illegible]	[illegible]	16.00	.0625
11.3	26	242.2	211.5	1155.8	[illegible]	[illegible]	15.42	.0649
12.3	27	244.3	213.7	1156.4	[illegible]	[illegible]	14.90	.0672
13.3	28	246.3	215.7	1157.1	[illegible]	[illegible]	14.40	.0696
14.3	29	248.3	217.8	1157.7	[illegible]	[illegible]	13.91	.0719
15.3	30	250.2	219.7	1158.3	[illegible]	[illegible]	13.50	.0742
16.3	31	252.1	221.6	1158.8	[illegible]	[illegible]	13.07	.0765
17.3	32	254.0	223.5	1159.4	[illegible]	[illegible]	12.68	.0788
18.3	33	255.7	225.3	1159.9	[illegible]	[illegible]	12.32	.0812
19.3	34	257.5	227.1	1160.5	[illegible]	[illegible]	12.00	.0835
20.3	35	259.2	228.8	1161.0	[illegible]	[illegible]	11.66	.0858
21.3	36	260.8	230.5	1161.5	[illegible]	[illegible]	11.36	.0880
22.3	37	262.5	232.1	1162.0	[illegible]	[illegible]	11.07	.0903
23.3	38	264.0	233.8	1162.5	[illegible]	[illegible]	10.80	.0926
24.3	39	265.6	235.4	1162.9	[illegible]	[illegible]	10.53	.0949
25.3	40	267.1	236.9	1163.4	[illegible]	[illegible]	10.28	.0972
26.3	41	268.6	238.5	1163.9	[illegible]	[illegible]	10.05	.0995
27.3	42	270.1	240.0	1164.3	[illegible]	[illegible]	9.83	.1018
28.3	43	271.5	241.4	1164.7	[illegible]	[illegible]	9.61	.1040
29.3	44	272.9	242.9	1165.2	[illegible]	[illegible]	9.41	.1063
30.3	45	274.3	244.3	1165.6	[illegible]	[illegible]	9.21	.1086
31.3	46	275.7	245.7	1166.0	[illegible]	[illegible]	9.02	.1108
32.3	47	277.0	247.0	1166.4	[illegible]	[illegible]	8.84	.1131
33.3	48	278.3	248.4	1166.8	[illegible]	[illegible]	8.67	.1153
34.3	49	279.6	249.7	1167.2	[illegible]	[illegible]	8.50	.1176
35.3	50	280.9	251.0	1167.6	[illegible]	[illegible]	8.34	.1198
36.3	51	282.1	252.2	1168.0	[illegible]	[illegible]	8.19	.1221
37.3	52	283.3	253.5	1168.4	[illegible]	[illegible]	8.04	.1243

TABLE 5—*Continued*

Gauge Pressure Lbs. per Sq. In.	Absolute Pressure Lbs. per Sq. In.	Temp. Degrees F.	Total Heat above 32° F.		Latent Heat H-h	Relative Volume	Cubic Feet in 1 Lb. Wt. of Steam	Wt. of 1 Cubic Foot of Steam, Lbs.
			In the Water h Heat-units	In the Steam H Heat-units	Heat-units			
38.3	53	284.5	254.7	1168.7	914.0	492	7.90	.1266
39.3	54	285.7	256.0	1169.1	913.1	484	7.76	.1288
40.3	55	286.9	257.2	1169.4	912.3	476	7.63	.1311
41.3	56	288.1	258.3	1169.8	911.5	468	7.50	.1333
42.3	57	289.1	259.5	1170.1	910.6	460	7.38	.1355
43.3	58	290.3	260.7	1170.5	909.8	453	7.26	.1377
44.3	59	291.4	261.8	1170.8	909.0	446	7.14	.1400
45.3	60	292.5	262.9	1171.2	908.2	439	7.03	.1422
46.3	61	293.6	264.0	1171.5	907.5	432	6.92	.1444
47.3	62	294.7	265.1	1171.8	906.7	425	6.82	.1466
48.3	63	295.7	266.2	1172.1	905.9	419	6.72	.1488
49.3	64	296.8	267.2	1172.4	905.2	413	6.62	.1511
50.3	65	297.8	268.3	1172.8	904.5	407	6.53	.1533
51.3	66	298.8	269.3	1173.1	903.7	401	6.43	.1555
52.3	67	299.8	270.4	1173.4	903.0	395	6.34	.1577
53 3	68	300.8	271.4	1173.7	902.3	390	6.25	.1599
54.3	69	301.8	272.4	1174.0	901.6	384	6.17	.1621
55.3	70	302.7	273.4	1174.3	900.9	379	6.09	.1643
56.3	71	303.7	274.4	1174.6	900.2	374	6.01	.1665
57.3	72	304.6	275.3	1174.8	899.5	369	5.93	.1687
58.3	73	305.6	276.3	1175.1	898.9	365	5.85	.1709
59.3	74	306.5	277.2	1175.4	898.2	360	5.78	.1731
60.3	75	307.4	278.2	1175.7	897.5	356	5.71	.1753
61.3	76	308.3	279.1	1176.0	896.9	351	5.63	.1775
62.3	77	309.2	280.0	1176.2	896.2	347	5.57	.1797
63.3	78	310.1	280.9	1176.5	895.6	343	5.50	.1819
64.3	79	310.9	281.8	1176.8	895.0	339	5.43	.1840
65.3	80	311.8	282.7	1177.0	894.3	334	5.37	.1862
66.3	81	312.7	283.6	1177.3	893.7	331	5.31	.1884
67.3	82	313.5	284.5	1177.6	893.1	327	5.25	.1906
68.3	83	314.4	285.3	1177.8	892.5	323	5.18	.1928
69.3	84	315.2	286.2	1178.1	891.9	320	5.13	.1950
70.3	85	316.0	287.0	1178.3	891.3	316	5.07	.1971
71.3	86	316.8	287.9	1178.6	890.7	313	5.02	.1993
72.3	87	317.7	288.7	1178.8	890.1	309	4.96	.2015
73.3	88	318.5	289.5	1179.1	889.5	306	4.91	.2036
74.3	89	319.3	290.4	1179.3	888.9	303	4.86	.2058
75.3	90	320.0	291.2	1179.6	888.4	299	4.81	.2080

TABLE 5—*Continued*

Gauge Pressure Lbs. per Sq. In.	Absolute Pressure Lbs. per Sq. In.	Temp. Degrees F.	Total Heat above 32° F.		Latent Heat L = Heat-units	Relative Volume	Cubic Feet in 1 Lb. Wt. of Steam	Wt. of 1 Cubic Foot of Steam, Lbs.
			In the Water h Heat-units	In the Steam H Heat-units				
76.3	91	320.8	292.0	1179.8	887.8	296	4.76	.2102
77.3	92	321.6	292.8	1180.0	887.2	293	4.71	.2123
78.3	93	322.4	293.6	1180.3	886.7	290	4.66	.2145
79.3	94	323.1	294.4	1180.5	886.1	287	4.62	.2166
80.3	95	323.9	295.1	1180.7	885.6	285	4.57	.2188
81.3	96	324.6	295.9	1181.0	885.0	282	4.53	.2210
82.3	97	325.4	296.7	1181.2	884.5	279	4.48	.2231
83.3	98	326.1	297.4	1181.4	884.0	276	4.44	.2253
84.3	99	326.8	298.2	1181.6	883.4	274	4.40	.2274
85.3	100	327.6	298.9	1181.8	882.9	271	4.36	.2296
86.3	101	328.3	299.7	1182.1	882.4	268	4.32	.2317
87.3	102	329.0	300.4	1182.3	881.9	266	4.28	.2339
88.3	103	329.7	301.1	1182.5	881.4	264	4.24	.2360
89.3	104	330.4	301.9	1182.7	880.8	261	4.20	.2382
90.3	105	331.1	302.6	1182.9	880.3	259	4.16	.2403
91.3	106	331.8	303.3	1183.1	879.8	257	4.12	.2425
92.3	107	332.5	304.0	1183.4	879.3	254	4.09	.2446
93.3	108	333.2	304.7	1183.6	878.8	252	4.05	.2467
94.3	109	333.9	305.4	1183.8	878.3	250	4.02	.2489
95.3	110	334.5	306.1	1184.0	877.9	248	3.98	.2510
96.3	111	335.2	306.8	1184.2	877.4	246	3.95	.2531
97.3	112	335.9	307.5	1184.4	876.9	244	3.92	.2553
98.3	113	336.5	308.2	1184.6	876.4	242	3.88	.2574
99.3	114	337.2	308.8	1184.8	875.9	240	3.85	.2596
100.3	115	337.8	309.5	1185.0	875.5	238	3.82	.2617
101.3	116	338.5	310.2	1185.2	875.0	236	3.79	.2638
102.3	117	339.1	310.8	1185.4	874.5	234	3.76	.2660
103.3	118	339.7	311.5	1185.6	874.1	232	3.73	.2681
104.3	119	340.4	312.1	1185.8	873.6	230	3.70	.2703
105.3	120	341.0	312.8	1185.9	873.2	228	3.67	.2724
106.3	121	341.6	313.4	1186.1	872.7	227	3.64	.2745
107.3	122	342.2	314.1	1186.3	872.3	225	3.62	.2766
108.3	123	342.9	314.7	1186.5	871.8	223	3.59	.2788
109.3	124	343.5	315.3	1186.7	871.4	221	3.56	.2809
110.3	125	344.1	316.0	1186.9	870.9	220	3.53	.2830
111.3	126	344.7	316.6	1187.1	870.5	218	3.51	2851
112.3	127	345.3	317.2	1187.3	870.0	216	3.48	.2872
113.3	128	345.9	317.8	1187.4	869.6	215	3.46	.2894

TABLE 5—*Continued*

Gauge Pressure Lbs. per Sq. In.	Absolute Pressure Lbs. per Sq. In.	Temp. Degrees F.	Total Heat Above 32° F.		Latent Heat H-h Heat-units	Relative Volume	Cubic Feet in 1 Lb. Wt. of Steam	Wt. of 1 Cubic Foot of Steam, Lbs.
			In the Water h Heat-units	In the Steam H Heat-units				
114.3	129	346.5	318.4	1187.6	869.2	213	3.43	.2915
115.3	130	347.1	319.1	1187.8	868.7	212	3.41	.2936
116.3	131	347.6	319.7	1188.0	868.3	210	3.38	.2957
117.3	132	348.2	320.3	1188.2	867.9	209	3.36	.2978
118.3	133	348.8	320.8	1188.3	867.5	207	3.33	.3000
119.3	134	349.4	321.5	1188.5	867.0	206	3.31	.3021
120.3	135	350.0	322.1	1188.7	866.6	204	3.29	.3042
121.3	136	350.5	322.6	1188.9	866.2	203	3.27	.3063
122.3	137	351.1	323.2	1189.0	865.8	201	3.24	.3084
123.3	138	351.8	323.8	1189.2	865.4	200	3.22	.3105
124.3	139	352.2	324.4	1189.4	865.0	199	3.20	.3126
125.3	140	352.8	325.0	1189.5	864.6	197	3.18	.3147
126.3	141	353.3	325.5	1189.7	864.2	196	3.16	.3169
127.3	142	353.9	326.1	1189.9	863.8	195	3.14	.3190
128.3	143	354.4	326.7	1190.0	863.4	193	3.11	.3211
129.3	144	355.0	327.2	1190.2	863.0	192	3.09	.3232
130.3	145	355.5	327.8	1190.4	862.6	191	3.07	.3253
131.3	146	356.0	328.4	1190.5	862.2	190	3.05	.3274
133.3	148	357.1	329.5	1190.9	861.4	187	3.02	.3316
135.3	150	358.2	330.6	1191.2	860.6	185	2.98	.3358
140.3	155	360.7	333.2	1192.0	858.7	179	2.89	.3463
145.3	160	363.3	335.9	1192.7	856.9	174	2.80	.3567
150.3	165	365.7	338.4	1193.5	855.1	169	2.72	.3671
155.3	170	368.2	340.9	1194.2	853.3	164	2.65	.3775
160.3	175	370.5	343.4	1194.9	851.6	160	2.58	.3879
165.3	180	372.8	345.8	1195.7	849.9	156	2.51	.3983
170.3	185	375.1	348.1	1196.3	848.2	152	2.45	.4087
175.3	190	377.3	350.4	1197.0	846.6	148	2.39	.4191
180.3	195	379.5	352.7	1197.7	845.0	144	2.33	.4296
185.3	200	381.6	354.9	1198.3	843.4	141	2.27	.4400
190.3	205	383.7	357.1	1199.0	841.9	138	2.22	.4503
195.3	210	385.7	359.2	1199.6	840.4	135	2.17	.4605
200.3	215	387.7	361.3	1200.2	838.9	132	2.12	.4707
205 3	220	389.7	362.2	1200.8	838.6	129	2.06	.4852
245.3	260	404.4	377.4	1205.3	827.9	110	1.76	.5686
285.3	300	417.4	390.9	1209.2	818.3	96	1.53	.6515
485.3	500	467.4	443.5	1224.5	781.0	59	.94	1 062
685.3	700	504.1	482.4	1235.7	753.3	42	.68	1.470
985.3	1000	546.8	528.3	1248.7	720.3	30	.48	2.082

Questions

1. What is combustion?
2. What is one of the main factors in combustion?
3. Of what is air composed?
4. In what proportion are these two gases combined?
5. What is the principal constituent of coal and other fuels?
6. What other valuable constituent is contained in bituminous coal?
7. What is the usual temperature of a boiler furnace when in active operation?
8. About what should be the temperature of the escaping gases?
9. What two factors are indispensable in the economical use of coal?
10. What is heat?
11. What is the heat unit?
12. What is the mechanical equivalent of heat?
13. How many heat units are there in one pound of carbon?
14. How many heat units are there in one pound of hydrogen gas?
15. What is specific heat?
16. What is sensible heat?
17. What is latent heat?
18. Is the latent heat imparted to a body lost?
19. What is meant by the total heat of evaporation?
20. How much heat expressed in heat units is required to evaporate one pound of water from a temperature of 32° into steam at atmospheric pressure?
21. Name the two elements composing pure water.
22. In what proportion are these two gases combined in the formation of water?

23. Is perfectly pure water desirable for use in steam boilers?

24. What causes scale to form in boilers?

25. What proportion of mineral matter is usually found in water?

26. What is steam?

27. Of what does matter consist?

28. How does the application of heat to any substance affect its molecules?

29. In what particular manner does heat affect the molecules of water?

30. Is steam under pressure visible?

31. What is saturated steam?

32. What is dry steam?

33. What is superheated steam?

34. What is meant by the term total heat in steam?

35. What is meant by the density of steam?

36. What is meant by the volume of steam?

37. What is the weight of a cubic foot of water at 39.1° temperature?

38. What is the weight of a cubic foot of water at a temperature of 212°?

39. What is the boiling point of water in the open air at sea level?

40. At what temperature will water boil in a perfect vacuum?

41. What is meant by the relative volume of steam?

Answers.

1. A union of the carbon and hydrogen of the fuel with the oxygen of the air. Each atom of carbon combines with two atoms of oxygen, the result of this combination being heat.

2. A proper supply of air.

3. Two gases: Oxygen and nitrogen.

4. By volume, 21 per cent of oxygen and 79 per cent of nitrogen; or by weight, 23 per cent of oxygen and 77 per cent of nitrogen.

5. Carbon.

6. Hydrogen.

7. 2500 degrees, sometimes reaching 3000 degrees F.

8. 400 degrees F. for bituminous coal.

9. A constant high temperature in the furnace and quick combustion.

10. A result due to an internal commotion or vibration of the molecules or atoms of which all matter consists.

11. The quantity of heat required to raise the temperature of one pound of water one degree, or from 39 degrees to 40 degrees F.

12. The energy required to lift 778 pounds one foot high, or 77.8 pounds 10 feet high, or 1 pound 778 feet high.

13. 14,500.

14. 62,000.

15. The specific heat of any substance is the ratio of the quantity of heat required to raise a given weight of

that substance one degree in temperature, to the quantity of heat required to raise an equal weight of water one degree in temperature, the water being at its maximum density, which is 39.1 degrees F. Specific heat of water is 1.

16. Heat that can be measured in degrees by a thermometer.

17. Heat that is not shown by the thermometer.

18. It is not. On the contrary while becoming latent it is at the same time doing much work.

19. The sum of the sensible and latent heat required to evaporate water into steam.

20. 1146 heat units.

21. The two gases: Hydrogen and oxygen.

22. By volume—Hydrogen 2, oxygen 1. By weight—Hydrogen 11.1, oxygen 88.9.

23. It is not.

24. Carbonates of lime and magnesia, also the sulphate of lime.

25. From ten to forty grains per gallon.

26. Water in its gaseous form, made so by the application of heat.

27. Molecules or atoms inconceivably small in size, yet each having an individuality, and in the case of solids and liquids, each having a mutual cohesion or attraction for the other, and all being in a state of continual vibration more or less violent according to the temperature of the mass.

28. It tends to accelerate the vibrations of the molecules. If enough heat is applied it will reduce the hardest substance to a liquid, or gaseous form.

29. It increases the natural vibrations of the molecules until they lose all attraction for each other and instead become entirely repulsive, and if not confined will fly off into space; but being confined, they continually strike against the sides of the containing vessel, thus causing the pressure which steam exerts when under confinement.

30. It is not.

31. Steam taken directly from the boiler without having been superheated.

32. Steam that contains no moisture. It may be either saturated or superheated.

33. Steam that has been conducted through a vessel, or coils of pipe, separate from the boiler in which it was generated, and is there heated to a higher temperature than that due to its pressure.

34. The heat required to raise the temperature of the water from 32 degrees F. to the temperature of the steam, plus the heat required to evaporate the water at that temperature. This latter heat becomes latent in the steam and is therefore called the latent heat of steam.

35. The actual weight in pounds or fractions of a pound avoirdupois of a given volume of steam, as one cubic foot. This should not be confounded with pounds pressure.

36. The number of cubic feet in one pound weight of steam.

37. Approximately 62.5 pounds.

38. 59.76 pounds.

39. 212 degrees F.

40. 32 degrees F.

41. The number of volumes of steam produced from one volume of water. For instance, at 100 lbs. gauge pressure the steam produced from one cubic foot of water will have a volume of about 238 cubic feet.

CHAPTER VII

EVAPORATION TESTS

Evaporation tests, object of—Preparing for a test—Suggestions as to apparatus needed—Taking the temperature of the feed water—Precautions necessary to obtain accurate results—Duration of test—Feeding a boiler during a test—How to proceed if the boiler is fed by an injector—Determining the percentage of moisture in the steam—Moisture in the coal—Chimney draft—Draft gauge—Rules for determining the results of a test—"Equivalent evaporation," how to compute—Factors of evaporation—Boiler horse power.

Evaporation Tests. The object of making evaporation tests of steam boilers is primarily to ascertain how many pounds of water the boilers are evaporating per pound of coal burned; but these tests can and should be made to determine several other important points with reference to the operation of the boilers, as for instance: 1. The efficiency of the boiler and furnace as an apparatus for the consumption of fuel and the evaporation of water; whether this apparatus is performing its guaranteed duty in this respect, and how it compares with a known standard. 2. To determine the relative economy of different varieties of coal, also to determine the relative value of fuels other than coal, such as oil, gas, etc. 3. To ascertain whether or not the boilers as they are operated under ordinary every day conditions are being run as economically as they should be. 4. In case the boilers, owing to an increased demand for steam, fail to supply a sufficient quantity without forcing the fires, whether or not additional boilers are needed, or whether the trouble could

be overcome by a change of conditions in operating them.

As was stated in the chapter on boiler setting, every steam plant can and should be equipped with the necessary apparatus for making evaporation tests, and every engineer should take pride in making a good showing in the economical use of coal, and, be it said to their credit, the majority of engineers do this, although many of them are working under conditions that prevent them from doing all that they might desire along this line. Too many engineers are compelled to look after work entirely outside of and foreign to their vocation as engineers, often having to go to some distant part of the building to make repairs to some part of the machinery, leaving their boiler and engine to care for themselves for the time being, thus not only endangering the safety of property, but of life as well. But conditions are gradually changing for the better in this respect, and employers and owners of steam plants are coming more and more to recognize the fact that the engineer is something more than a mere handy man about a factory, in fact, that he has a distinct and responsible vocation to be fulfilled, viz., the safe and economical operation of the plant where the power comes from.

The author proposes to present in as brief a manner as possible a few simple suggestions and rules for the benefit of engineers who desire to make evaporation tests with a view of determining one or more of the points mentioned at the beginning of this chapter.

Tests for the last three purposes named can be made by the regular engineering force of the plant, but in case a controversy should arise between the maker of the boiler and the purchaser regarding the first men-

tioned point, viz., the guaranteed efficiency of the boiler or the furnace, the services of experts in boiler testing may be resorted to.

Preparing for a Test. All testing apparatus should be kept in such shape that it will not take three or four days to get it ready for making a test. On the contrary, it can be and should be always kept in condition ready for use, so that the preparations for making a test will occupy but a short time. A small platform scale sufficiently large for weighing a wheel-barrow load of coal should also be provided in addition to the apparatus referred to in Chapter II.

The capacity of each of the two tanks therein mentioned, and which are illustrated in Fig. 10, can be determined in two ways, either by measuring the cubical contents of each or by placing them one at a time on the scales, filling them with water to within a few inches of the top, and note the weight. Also make a permanent mark on the inside at the water level. The water should then be permitted to run out until within an inch or so of the outlet pipe near the bottom, where another plain mark should be made, after which the empty tank should be again weighed, then by subtracting the last weight from the first the exact number of pounds of water that the tank will contain between the top and bottom marks can be determined and a note made of it.

It is much more convenient to have each tank contain the same quantity of water, although not absolutely necessary. The tanks should also be numbered 1 and 2 respectively in order to prevent confusion in keeping a record of the number of tanks full of water used during the test. Care should be exercised to have the water with which the tanks are filled while on

the scale, at or near the same temperature as that at which it is to be fed into the boiler during the test. Otherwise there is liability of error owing to the variation in the weight of water at different temperatures. In order to guard against this, the capacity in cubic feet of each tank between the top and bottom marks should be ascertained by measuring the distance between the marks, also the diameter, or, if the tanks be square, the length of one side, after which the cubical contents can be easily figured and noted down. By knowing the capacity in cubic feet of each tank all possibility of error in the weight of feed water will be eliminated.

The scales for weighing the coal can be fitted with a temporary wooden platform large enough to accommodate a wheel-barrow, and after it has been balanced with the empty barrow on it, the record of weight of coal burned during the test can be easily kept.

The same barrow should be used throughout the test, and to save complications in estimating the weight, the same number of pounds of coal should be filled in the barrow each load. The coal passer will learn in a short time to fill the barrow to within a few pounds of the same weight each load by counting the shovelsful and the difference can easily be adjusted by having a small box of coal near the scale from which to take a few lumps to balance the load, or if there is too much coal in the barrow some of it can be thrown into the box.

At least two separate tally sheets should be provided, marked respectively coal and water, and the one for coal placed near the scale, and care should be taken that each load is tallied as soon as it is weighed. The tally sheet for water should be near the measuring

tanks and as soon as a tank is emptied it should be tallied. The temperature of the feed water should be taken at least every thirty minutes, or oftener if possible, from a thermometer placed in the feed pipe near the check valve, as described in Chapter II. The readings should be noted and, at the expiration of the test, the average taken.

If the object of the test is to ascertain the efficiency of the boiler and furnace it is absolutely necessary that the boiler and all its appurtenances be put in good condition, by cleaning the heating surface both inside the boiler and outside, scraping and blowing the soot out of the tubes if it be a return tubular boiler, and blowing the soot and ashes from between the tubes if it is a water tube boiler. All dust, soot and ashes should be removed from the outside of the shell and also from the combustion chamber and smoke connections. The grate bars and sides of the furnace should be cleared of all clinker, and all air leaks made as close as possible. The boiler and all its water connections should be free from leaks, especially the blow-off valve or cock. If any doubt exists as to the latter it should be plugged or a blind flange put on it. It is very essential that there should be no way for the water to leak out of the boiler, neither should any water be allowed to get into the boiler during the test except that which is measured by passing through the tanks.

The engineer making the test should know the number of square feet of grate surface and also the area of the heating surface in square feet. Rules for computing the latter are given in Chapter II. If the boiler is a water tube boiler the outside diameter of the tubes must be used in estimating the heating

surface. A correct knowledge of the above points is essential in making a test for determining any one of the four objects mentioned at the commencement of this chapter, but especially is it needed in conducting a test of the efficiency of the boiler and furnace.

In making an efficiency test it is essential that the boiler should be run at its fullest capacity from the beginning to the end of the test. Therefore arrangements should be made to dispose of the steam as fast as it is generated. If the boiler is in a battery connected with a common header, its mates can be fired lighter during the test, but if there is but the one boiler in use a waste steam pipe should be temporarily connected through which the surplus steam, if there is any, can be discharged into the open air through a valve regulated as required. Before starting the test the boiler should be thoroughly heated by having been run several hours at the ordinary rate. The fire should then be cleaned and put in good condition to receive fresh coal.

At the time of beginning the test the water level should be at or near the height ordinarily carried and its position marked by tying a cord around one of the guard rods of the gauge glass, and, to prevent all possibility of error, the height of the water in the glass should be measured and a note made of it. Note also the time that the first lot of weighed coal is fed to the furnace and record it as the starting time. The steam pressure should be noted at the beginning of the test and at regular intervals during the progress of the test in order that the average pressure may be obtained.

At the close of the test all of the above conditions should be as nearly as possible the same as at the

beginning; the quantity and condition of the fire should be the same, also the steam pressure and water level. This can be accomplished only by careful work towards the close of the test.

During the progress of the test care should be exercised to prevent any waste of coal, especially in cleaning the fire. The ash made during the test must not be wet down until after it is weighed, as in all calculations for combustible and non-combustible matter in the coal the ash should be dry.

The duration of the test should be at least ten hours if it is possible to continue it for that length of time. The feed pump should be kept running at such speed as will supply the water to the boiler as fast as it is evaporated, and no faster. If at the close of the test a portion of water is left in the last tank tallied it can be measured and deducted from the total. And if any weighed coal is left on the floor it should be weighed back and deducted from the total weight. If the boiler is fed by an injector instead of a pump during the test, the injector should receive steam directly from the boiler under test through a well protected pipe. Also, the temperature of the feed water should be taken from the measuring tanks, or at least from the suction side of the injector, for the reason that the water in passing through the injector receives a large quantity of heat imparted to it by live steam directly from the boiler. Therefore the temperature of the water after it leaves the injector would not be a true factor for figuring the evaporation.

Determination of the Percentage of Moisture in the Steam. This is an important point in estimating the results of an evaporation test for the reason that each pound weight of moisture in the steam as it leaves the

boiler represents a pound of water that has not been evaporated into steam, and should therefore be deducted from the total weight of water fed into the boiler during the test.

The steam should be tested for moisture by taking samples of it from the steam pipe or header as near the boiler as possible in order to guard against additional moisture caused by condensation. For making expert boiler tests for scientific and other purposes, an instrument called a calorimeter is used for ascertaining the quantity of moisture in the steam. But most engineers are not so fortunate as to possess one of these instruments, nor even to induce their employers to purchase one, therefore space will not be taken up in describing it here. A method of testing the quality of the steam by noting the appearance of a small jet blowing into the atmosphere is mentioned near the close of Chapter IV, which in the absence of a calorimeter would answer very well for ordinary purposes.

Moisture in the Coal. This can generally be obtained from the reports of the geologist of the state in which the coal is mined or from the dealer, although the former is the most reliable. The percentage of moisture must be deducted from the total weight of coal in figuring the weight of combustible.

Measuring the Chimney Draft. A good draft is indispensable for obtaining economical results in an evaporation test. The draft can be easily regulated by a damper to suit the conditions. Chimney draft is ordinarily measured by a draft gauge connected with the smoke flue near the chimney. The usual form of draft gauge is a glass tube bent in the shape of the letter U. (See Fig. 12.) One leg is connected to the flue by a

small rubber hose, while the other is open to the atmosphere. The tube is partly filled with water, which will, when there is no draft, stand at the same height in both legs. When connected to the chimney or flue the suction will cause the water in the leg to which the hose is attached to rise, while the level of the water in the other leg will be equally depressed, and the extent of the variation in fractions of an inch is the measure of the draft. Thus the draft is referred to as being .5, .7 or .75 inches. The draft should not be less than .5 inches in any case to insure good results.

FIGURE 12.

Having thus successfully conducted the test to its close, and being armed with all the data heretofore noted, the engineer is now ready to compute the results.

If the test is made for the purpose of determining the efficiency of the boiler and setting as a whole, including grate, chimney draft, etc., then the result must be based upon the number of pounds of water evaporated per pound of coal. This latter phrase includes not only the purely combustible matter in the coal, but the non-combustible also, as ash, moisture, etc. Some varieties of western coal contain as high as 12 to 14 per cent. of moisture, and the ability of the furnace

to extract heat from the mass is to be tested, as well as the ability of the boiler to absorb and transmit that heat to the water. Therefore the efficiency of the boiler and furnace =

$$\frac{\text{Heat absorbed per pound of coal.}}{\text{Heating value of one pound of coal.}}$$

If the test is to determine the efficiency of the boiler itself as an absorber of heat, then the combustible alone must be considered in working out the final result. Thus,

Efficiency of boiler =

$$\frac{\text{Heat absorbed per pound of combustible.}}{\text{Heating value of one pound of combustible.}}$$

When making a series of tests for the purpose of comparing the economical value of different varieties of coal the conditions should be as nearly uniform as possible; that is, let the tests be made under ordinary working conditions and with the same boiler or boilers, and if possible with the same fireman.

The following is a record of one of many evaporation tests made by the author, and is introduced here for the purpose of illustrating methods of computing the results to be obtained from the various data. The rather large quantity of coal burned per square foot of grate surface per hour (25 lbs.) is owing to the fact that the boiler was run to its full capacity, the coal burning clean and forming no clinker. The chimney draft also was exceptionally good, giving a large unit of evaporation per square foot of heating surface per hour. The low temperature of the escaping gases is due to the fact that they were returned over the top of the boiler before passing to the chimney.

Date of test				
Duration of test			12	hours.
Boiler, return tubular, 72 in. diameter, 18 ft. long, 72-4½ in. tubes.				
Kind of coal, Pocahontas, average steam pressure			85	lbs.
Weight of coal consumed			11,100	"
Weight of water apparently evaporated			107,157	"
Weight of dry ash returned			[illegible]	"
Moisture in the coal	2.0	" =	222	"
Moisture in the steam	1.0	" =	1,071	"
Dry coal corrected for moisture			10,878	"
Weight of combustible			9,978	"
Water corrected for moisture in the steam			106,116	"
Water evaporated into dry steam, from and at 212			[illegible]	"
Water evaporated per lb. of coal, actual conditions			9.65	"
Water evaporated per lb. of coal, from and at 212			10.61	"
Water evaporated per lb. of combustible, from and at 212			11.81	"
Water evaporated per lb. of dry coal, from and at 212			10.82	"
Water evaporated per hr. per sq. ft. of heating surface			6.22	"
Coal burned per sq. ft. of grate surface per hour			25	"
Horse power developed by boiler during test			284.5	
Temperature of feed water, average			141°	
Temperature of chimney gases, average			400	
Square feet of grate surface			36	
Square feet of heating surface			1576	
Ratio of grate surface to heating surface			43.7	

The results obtained will be taken up in their regular order beginning with, first, water evaporated into dry steam from and at 212°. As it may be of benefit to some, a short definition of the meaning of the above expression is here given.

The term "equivalent evaporation," or the evaporation from and at 212°, assumes that the feed water enters the boiler at a temperature of 212° and is evaporated into steam at 212° temperature and at atmospheric pressure. As for instance, if the top man hole plate were left out or some other large opening in the steam space allowed the steam to escape into the atmosphere as fast as it was generated. Owing to the variation in the temperatures of the feed water used in different tests, and also the variation in the steam pressure, it is absolutely necessary that the

results of all tests be brought by computation to the common basis of 212° in order to obtain a just comparison.

The process by which this is done is as follows: Referring to the record of the test it is seen that the steam pressure average was 85 lbs. gauge pressure, or 100 lbs. absolute, and that the temperature of the feed water was 141°. Referring again to Table No. 5, physical properties of steam, it will be seen that in a pound of steam at 100 lbs. absolute pressure there are 1,181.8 heat units, and in a pound of water at 141° temperature there are 109.9 heat units. It therefore took 1181.8 − 109.9 = 1071.9 heat units to convert one pound of feed water at 141° into steam at 85 lbs. pressure. To convert a pound of water at 212° into steam at atmospheric pressure, and 212° temperature requires 965.7 heat units, and the 1,071.9 heat units would evaporate 1,071.9 ÷ 965.7 = 1.11 lbs. water from and at 212°. The 1.11 is the factor of evaporation for 85 lbs. gauge pressure and 141° temperature of feed water, and by multiplying "water corrected for moisture in the steam" (see record of test), 106,116 lbs., by 1.11, the weight of water which could have been evaporated into steam from and at 212°, is obtained, which is 117,788 lbs. The factor of evaporation is based upon the steam pressure and the temperature of the feed water in any test and the formula for ascertaining it is as follows: Factor $= \frac{H - h}{965.7}$, in which H = total heat in the steam, and h = total heat in the feed water. It is used in shortening the process of finding the evaporation from and at 212°, and Table No. 6 gives the factor of evaporation for various pressures and temperatures.

Table 6

Factors of Evaporation

Feed Water Temperature	Gauge Press. 50 lbs.	Gauge Press. 60 lbs.	Gauge Press. 70 lbs.	Gauge Press. 80 lbs.	Gauge Press. 90 lbs.	Gauge Press. 100 lbs.	Gauge Press. 110 lbs.	Gauge Press. 120 lbs.	Gauge Press. 130 lbs.
212°	1.027	1.030	1.032	1.035	1.037	1.039	1.041	1.043	1.047
200°	1.039	1.042	1.045	1.047	1.050	1.052	1.054	1.056	1.059
191°	1.049	1.052	1.054	1.057	1.059	1.061	1.063	1.065	1.069
182°	1.058	1.061	1.064	1.066	1.069	1.071	1.073	1.075	1.078
173°	1.067	1.070	1.073	1.076	1.078	1.080	1.082	1.084	1.087
164°	1.077	1.080	1.083	1.085	1.087	1.090	1.091	1.093	1.097
152°	1.089	1.092	1.095	1.098	1.100	1.102	1.104	1.106	1.109
143°	1.099	1.102	1.105	1.107	1.109	1.111	1.113	1.115	1.119
134°	1.108	1.111	1.114	1.116	1.119	1.121	1.123	1.125	1.128
125°	1.118	1.121	1.123	1.126	1.128	1.130	1.132	1.134	1.137
113°	1.130	1.133	1.136	1.138	1.140	1.143	1.145	1.146	1.150
104°	1.138	1.142	1.145	1.148	1.150	1.152	1.154	1.156	1.159
95°	1.149	1.152	1.154	1.157	1.159	1.161	1.163	1.165	1.169
86°	1.158	1.161	1.164	1.166	1.169	1.171	1.173	1.174	1.178
77°	1.167	1.170	1.173	1.176	1.178	1.180	1.182	1.184	1.187
65°	1.180	1.183	1.186	1.188	1.190	1.192	1.194	1.196	1.200
56°	1.189	1.192	1.195	1.197	1.200	1.202	1.204	1.206	1.209
47°	1.199	1.201	1.204	1.207	1.210	1.211	1.213	1.215	1.218
38°	1.208	1.211	1.214	1.216	1.218	1.220	1.222	1.224	1.228

Second, water evaporated per pound of coal actual conditions = water apparently evaporated divided by coal consumed = 9.65 lbs. No accurate estimate regarding the quality of the coal or the efficiency of the boiler can be made from this figure (9.65 lbs.). It can be used, however, in estimating the cost of fuel for generating the steam; as, for instance, if the boiler is supplying steam to an engine that uses 30 lbs. of steam per horse power per hour, it will require 30 ÷ 9.65 = 3.1 lbs. of coal per horse power per hour; the "actual conditions" under which the boiler is being operated being the pressure of steam required by the engine and the temperature of the feed water.

Third, water evaporated per pound of coal from and at 212° = water evaporated into dry steam from and at 212° divided by coal consumed = 10.61 lbs. This figure is the proper one to use in comparing the relative economic values of different varieties of coal tested with the same boiler or boilers.

Fourth, water evaporated per pound of combustible from and at 212° = water evaporated into dry steam from and at 212° divided by weight of combustible = 11.81 lbs. This result is the one to be used for ascertaining the efficiency of the boiler, and the percentage of efficiency is found by dividing the heat absorbed by the boiler per pound of combustible by the heat value of one pound of combustible. The average heat value of bituminous and semi-bituminous coals is not far from 15,000 heat units per pound of combustible. In the evaporation of 11.81 pounds of water from and at 212° the heat absorbed was $11.81 \times 965.7 = 11,404.9$ heat units. The efficiency of the boiler therefore was $\frac{11,404.9 \times 100}{15,000} = 76$ per cent.

In like manner to ascertain the efficiency of the boiler and furnace as a whole, the water evaporated from and at 212° per pound of coal is taken. Thus, $10.61 \times 965.7 = 10,246$ heat units absorbed from each pound of coal. Now assuming that there were 13,500 heat units in each pound of the coal used in the test, the per cent of efficiency of boiler and furnace was

$$\frac{10,246 \times 100}{13,500} = 75.9.$$

Fifth, water evaporated per pound of dry coal from and at 212° = water evaporated into dry steam from and at 212° divided by coal corrected for moisture. Thus, $117,788 \div 10,878 = 10.82$ lbs. This result is useful for calculating the results of tests of the same grade

of coal, but differing in the degree of moisture in each.

Sixth. Boiler horse power. The latest decision of the American Society of Mechanical Engineers (than whom there is no better authority) regarding the horse power of a boiler is as follows: "The unit of commercial horse power developed by a boiler shall be taken as 34½ units of evaporation per hour. That is, 34½ lbs. of water evaporated per hour from a feed temperature of 212° into steam of the same temperature. This standard is equivalent to 33,317 B. T. U. per hour. It is also practically equivalent to an evaporation of 30 lbs. of water from a feed water temperature of 100° F. into steam of 70 lbs. gauge pressure."

According to this rule the horse power developed by the boiler during the test under consideration = water evaporated into dry steam from and at 212°, 117,788 lbs ÷ 12 hrs. ÷ 34.5 = 284.5 horse power.

Questions

1. What is the primary object of an evaporation test?
2. Name four other important points which can be determined by evaporation tests.
3. In making a test of the efficiency of the boiler and furnace, what precautions should be observed?
4. How is the heating surface of water tube boilers estimated?
5. What length of time should a test be conducted?
6. In case the boiler is fed by an injector, what precautions are necessary?
7. If the steam contains any moisture, what should be done?
8. How is the weight of combustible determined?

9. What is a calorimeter, and for what purpose is it used?

10. By what other method may the moisture in the steam be estimated approximately?

11. What should be done with the percentage of moisture in the coal?

12. How is the chimney draft measured?

13. Describe a draft gauge.

14. Give the formula for ascertaining the efficiency of the boiler and setting.

15. If the test is to determine the efficiency of the boiler alone, what factors are used?

16. If a series of tests is made for comparing different varieties of coal, what should be done?

17. What is meant by equivalent evaporation?

18. Why should the results of all tests be computed from and at 212°?

19. What is a factor of evaporation?

20. How is it determined?

21. What is a boiler horse power?

22. How many heat units per hour is this equivalent to?

ANSWERS

1. To ascertain how many pounds of water the boiler is evaporating per pound of coal burned.

2. First, the efficiency of the boiler and furnace. Second, to determine the relative economy of different kinds of coal and other fuels. Third, to ascertain if the boiler is being operated economically. Fourth, in case the boiler fails to supply sufficient steam without forcing the fire, whether or not the trouble could be remedied by a change of operating conditions.

3. The boiler should be in good condition, and no

water fed to the boiler during the test except that by passing through the measuring tanks; neither should any fuel be passed into the furnace except that which is weighed.

4. By the same method as that used in the case of tubular boilers, except that the outside diameters of the tubes are used in the calculation.

5. Ten hours at the least.

6. Steam for the injector should be taken directly from the boiler under test. The temperature of the feed water should be taken at the measuring tanks before it enters the injector.

7. It should be tested for moisture by means of the calorimeter.

8. The weight of the dry ashes made during the test, also the percentage of moisture in coal, should be deducted from the total weight of coal consumed.

9. An instrument consisting of a small reservoir fitted to receive one or more thermometers, and in some types a pressure gauge. The device is connected to the steam pipe by means of a small size nipple and valve through which samples of the steam may, at stated intervals, be admitted to the reservoir and there tested for moisture.

10. Note the appearance of a small jet blowing into the atmosphere. Dry steam will be transparent as it leaves the opening, while steam containing too much moisture will appear as vapor immediately upon leaving the opening.

11. It should be deducted from the total weight of coal when figuring weight of combustible.

12. By means of a draft gauge.

13. A glass tube bent in the shape of the letter U. One leg is connected to the smoke flue while the other leg is open to the atmosphere. The tube is partly

filled with water which will, when there is no draft, stand at the same height in both legs, but with a draft in the flue the water will rise in the connected leg and be equally depressed in the opposite leg, the extend of the variations in fractions of an inch being the measure of the draft.

14. Heat absorbed per pound of coal, divided by heating value of one pound of coal.

15. Heat absorbed per pound of combustible, divided by heating value of one pound of combustible.

16. The tests should be made under ordinary working conditions, with the same boiler or boilers, and if possible with the same fireman.

17. The term assumes that the feed water enters the boiler at a temperature of 212 degrees F. and is evaporated into steam at 212 degrees and at atmospheric pressure.

18. Owing to the variation in the temperatures of the feed water used in different tests, and also the variation in the steam pressure.

19. A factor based upon the steam pressure and the temperature of the feed water. It is used in calculations of equivalent evaporation.

20. From the total number of heat units in the steam subtract the number of heat units in the feed water and divide the remainder by the constant number 965.7. Result will be the factor of evaporation for that particular test.

21. 34½ pounds of water evaporated per hour from a feed temperature of 212 degrees F. into steam of the same temperature. This is practically equivalent to an evaporation of 30 pounds of water per hour from a feed temperature of the feed water.

22. 33,317.

CHAPTER VIII

MECHANICAL STOKERS

Principles involved in the action of automatic stokers—Advantages and disadvantages attending their use—Classification and general description of stokers—Coal handling machinery—Under feed stokers—Mansfield chain grate stoker—Playford stoker—Vicars mechanical stoker—The Wilkinson stoker—Murphy stoker—Roney stoker—The American under-feed stoker—The Jones under-feed stoker—Outside furnaces—Conditions required in a boiler furnace to ensure good combustion—Hindrances to good combustion—Description of Burke outside furnace.

The principles governing the operation of mechanical or automatic stokers are in the main correct, viz., that the supply of coal and air is continuous and that provision is made for the regulation of the supply of fuel according to the demand upon the boiler for steam; also that the intermittent opening and closing of the furnace doors, as in hand firing, thereby admitting a large volume of cold air directly into the furnace on top of the fire, is avoided.

Mechanical stokers have within the last twelve years been largely adopted in the United States, especially in sections where bituminous coal is the principal fuel. The disadvantages attending their use are:

First, that the cost of properly installing them is so great that their use is practically prohibited to the small manufacturer.

Second, that in case of a sudden demand upon the boilers for more steam the automatic stoker cannot respond as promptly as in hand firing, although

this objection could no doubt be met by skillful handling.

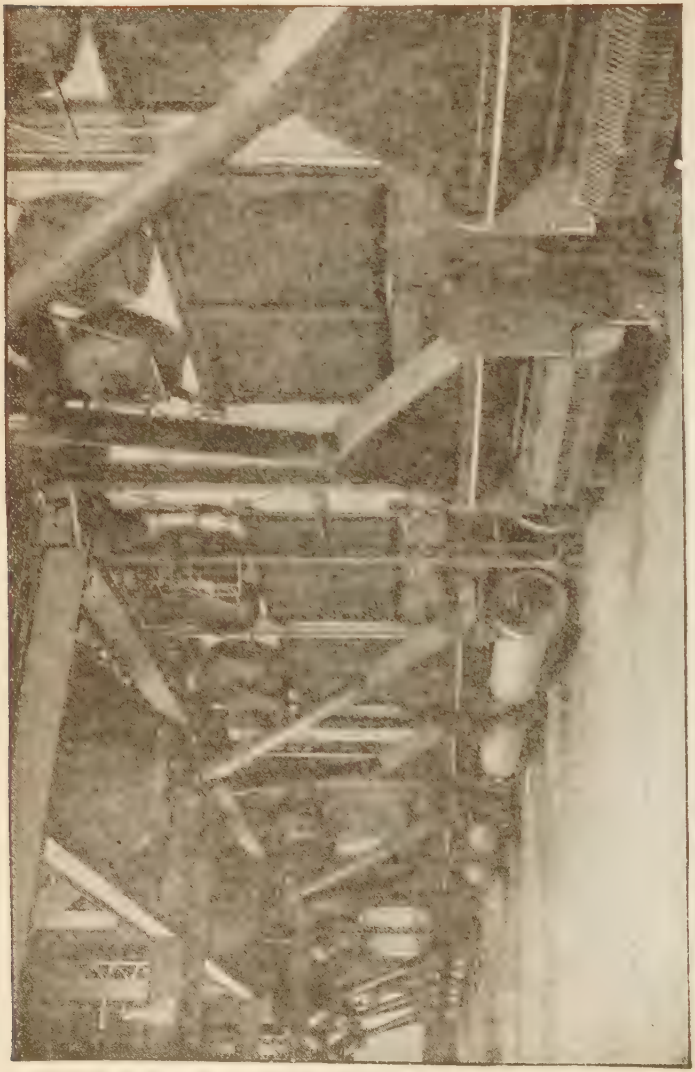
Third, the extra cost for power to operate them, although this is probably offset by the diminished expense for labor required, as compared to hand firing.

There are many different types of mechanical stokers and automatic furnaces, but they may for convenience be grouped into four general classes. In class one the grate consists of an endless chain of short bars that travel in a horizontal direction over sprocket wheels, operated either by a small auxiliary engine or by power derived from an overhead line of shafting in front of the boilers.

In class two may be included stokers having grate bars somewhat after the ordinary type as to length and size, but having a continuous motion up and down or forward and back. This motion, though slight, serves to keep the fuel stirred and loosened, thus preventing the firing from becoming sluggish. The grate bars in this class of stokers are either horizontal or inclined at a slight angle, and their constant motion tends to gradually advance the coal from the front to the back end of the furnace.

Class three includes stokers having the grate bars steeply inclined. Slow motion is imparted to the grates, the coal being fed onto the upper end and forced forward as fast as required.

Class four includes an entirely different type of stoker, in that the fresh coal is supplied underneath the grates, and is pushed up through an opening left for the purpose midway of the furnace. The gases, on being distilled, immediately come in contact with the hot bed of coke on top and the result is good combustion. In this type of stoker steam is the active agent used for forcing the coal up into the furnace, either



by means of a long, slowly-revolving screw, as in the American stoker, or a steam ram, as with the Jones under-feed stoker. A forced draft is employed, and the air is blown into the furnace through tuyeres When these stokers are intelligently handled they give good results, especially with cheap bituminous coal. The clinker formed on the grate bars or dead plates is easily removed.

The coal is supplied to mechanical stokers either by being shoveled by hand into hoppers in front of and above the grates, or, as is the case in most of the large plants using them, it is elevated by machinery and deposited in chutes, through which it is fed to each boiler by gravity. Stokers of the chain grate variety are usually constructed so that they may be withdrawn from underneath the boiler in case repairs are necessary. The coal, either nut or screenings, is fed into a hopper in front of and above the level of the grate, and is slowly carried along towards the rear end. The ash drops from the grate as it passes over the sprocket wheel at the rear.

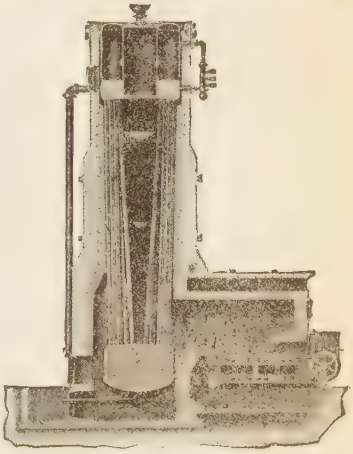


FIGURE 112.
CAHALL VERTICAL BOILER WITH CHAIN GRATE STOKER ATTACHED

Fig. 111 shows a battery of Babcock and Wilcox water tube boilers fitted with chain grate stokers. The

buckets for elevating the coal to bins overhead, from whence it is fed by gravity to the stokers, are not shown. These buckets or carriers may also be utilized for conveying the ashes from the boiler-room.

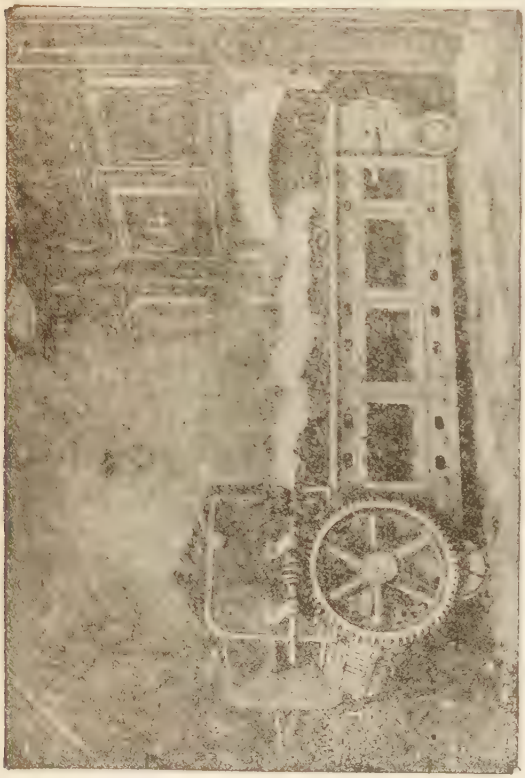


FIGURE 113.
MANSFIELD CHAIN GRATE STOKER, SHOWING HOW IT CAN BE WITHDRAWN FROM UNDER BOILER.

Fig. 112 is a sectional view of a vertical Cahall boiler with a Mansfield chain grate stoker, and Fig. 113 shows the same stoker withdrawn from the boiler.

The Coxe mechanical stoker operates upon the same

general principles as do those previously described, being of the chain grate type, but it has in addition a series of air chambers just underneath the upper traveling grate. These air chambers are made of sheet iron and are open at the top and provided with dampers for regulating the air pressure for different sections of the grate. The air blast is supplied by a fan. Another feature of this stoker is a water chamber for the bottom section of the grate to travel through on its return.

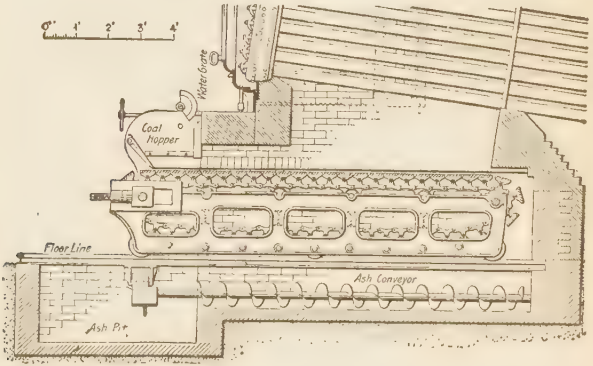


FIGURE 114.

The Playford stoker has wrought iron T bars extending across the furnace and attached to the traveling chains. These T bars carry the small cast iron sections composing the grate. A screw conveyor is also placed in the ash pit for the purpose of carrying the ashes from the rear to the front of the pit. Fig. 114 is a sectional view of the Playford stoker attached to a water tube boiler.

In class two may be included stokers having the grates inclined more or less. In some varieties the grates incline from front to rear, while in others they

are made to incline from the side walls towards the center line of the furnace.

In the Vicars mechanical stoker the grate bars are somewhat of the shape of the ordinary grate and lie in two tiers in a horizontal position. The lower or back tier next the bridge wall is stationary and is placed there for the purpose of catching what coal is carried over the ends of the upper or moving grate bars. These have a slow reciprocating motion which gradually moves the hot coke back towards the bridge wall. The coal is fed from a hopper into two compartments, from which it is pushed by reciprocating plungers onto a coking plate and from thence it passes to the grate bars. The motion of these bars has several intermediate variations, from a state of rest to a movement of $3\frac{1}{2}$ in. They have a simultaneous movement forward by which the fuel is advanced, but on the return movement the bars act at separate intervals. In this manner the fuel remains undisturbed by the return motion of the grates. Fig. 115 illustrates this stoker.

In the Wilkinson stoker, Fig. 116, the grate bars are hollow and are set at an angle of 20°, the inclination being from front to back. Each bar is stepped along its fire surface and on the rise is perforated with a long, narrow slot. A steam pipe extends along the front of the furnace and from this pipe small branch pipes lead into the ends of the grate bars, which latter are in fact a series of hollow trunks with their front ends open. When in operation a steam blast is admitted to each of the several trunk grate bars through the small branch pipes, and this blast induces an air current of more or less pressure, which finds an outlet through the narrow slots in the stepped fire

surface of the grates and directly into and through the burning mass of fuel. A slow reciprocating motion is imparted to the grates by means of cranks and links operated from an overhead shaft; see Fig. 117. These cranks are set alternately at 90° with each other, thus giving a forward movement to one-half of the grate bars while at the same time the other half is moving backward. In this manner the fuel is kept slowly moving down the inclined grates.

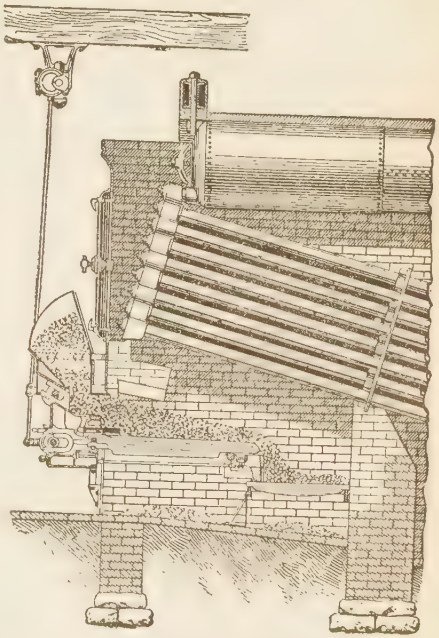


FIGURE 115.

The Murphy Automatic Furnace, a sectional view of which is shown in Fig. 118, has the grates inclined inwards from the side walls, while a fire brick arch is sprung from side to side to cover the entire length of the grate. The coal is shoveled or fed by carrier into the coal magazines, one on each side, as shown in the cut. If the furnace is placed directly under the boiler it necessitates putting these coal magazines within the side walls, but as the Murphy furnace is usually con-

structed at the present day as an outside furnace, the coal magazines are independent of the boiler walls.

The bottom of each magazine is used as a coking

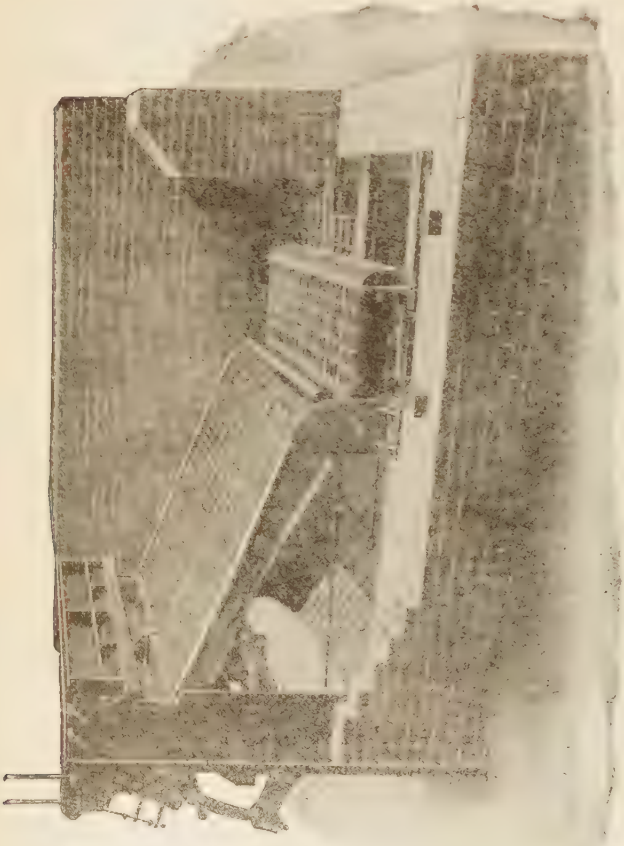


FIGURE 116.
FURNACE VIEW, WILKINSON STOKER.

plate, against which the upper ends of the inclined grates rest. On the central part of this plate is an inverted open box. This is termed the "stoker box,"

and it is moved back and forth across the face of the coking plate by means of a shaft with pinions that mesh into racks under each end of the box. By means

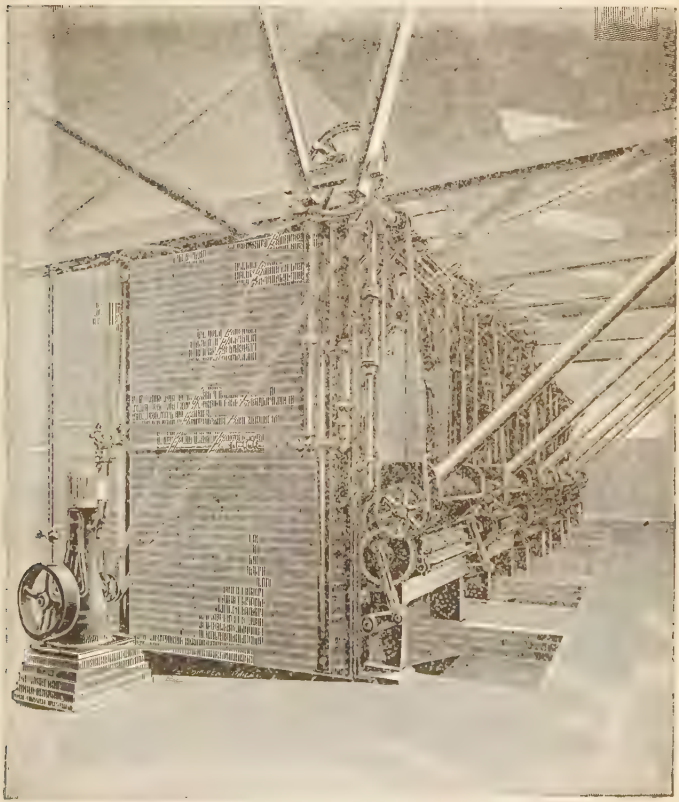


FIGURE 117. WILKINSON STOKER.

of this motion of the stoker boxes the coal is pushed forward to the edge of the coking plate and from thence it slowly passes down over the inclined grates

toward the center of the furnace. At this point the slowly rotating clinker breaker grinds the clinker and other refuse and deposits them in the ash pit.

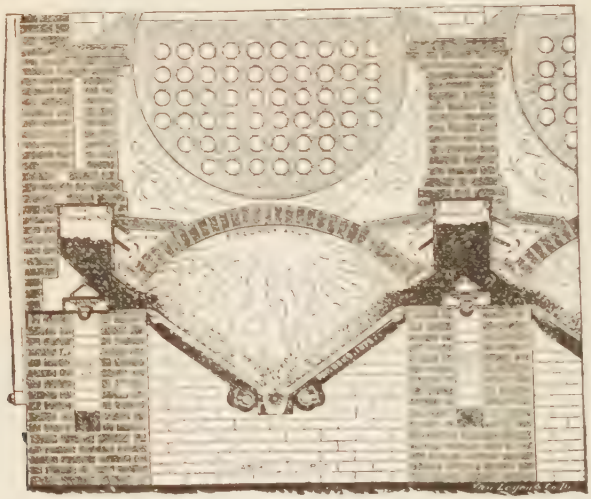


FIGURE 118. THE MURPHY AUTOMATIC FURNACE.

Above the coking plates are the "arch plates," upon which the bases of the fire brick arch rest. These plates are ribbed, the ribs being an inch apart, and, the arch resting upon these ribs, there is thus provided a series of air ducts by means of which the air, already heated by having been admitted in front and passed through the flues over the arch, is conducted into the furnace above the grates and comes directly in contact with the gases rising from the coking fuel. Air is also admitted under the coking plate and, passing up through the grates, serves to keep them cool and also furnishes the needed supply to the burning coke as it slowly moves down toward the center.

The fuel is aided in its downward movement by the constant motion of the grates, one grate of each pair being moved up and down by a rocker at the lower end.

Motion is imparted to the various moving parts of this furnace by means of a reciprocating bar extending across the outside of the entire front, and to which all the working parts are attached by links and levers. This bar is operated by a small engine at one side of the setting, the power required being about one horse-power per furnace.

The Roney stoker consists of a set of rocking stepped grate bars, inclined from the front towards the bridge wall. The angle of inclination is 37°. A dumping grate operated by hand is at the bottom of the incline for the purpose of receiving and discharging the clinker and ash. This dumping grate is divided into sections for convenience in handling.

The coal is fed onto the inclined grates from a hopper in front. The grate bars rock through an arc of 30°, assuming alternately the stepped and the inclined position. Fig. 119 is a sectional perspective view of this stoker and illustrates the working parts.

The grate bars receive their motion through the medium of a rocker bar and connecting rod. A shaft extending across the front of the stoker under the coal hopper carries an eccentric that gives motion to the connecting rod and also to the pusher in the coal hopper. This pusher, working back and forth, feeds the coal over the dead plate onto the grates, and its range of motion is regulated by a feed wheel from no stroke to full stroke, according to the demand for coal. The motion of the grate bars may also be regulated by

a sheath nut working on a long thread on the connecting rod. Each grate bar consists of two parts, viz., a cast iron web fitted with trunnions on each end that rest in seats in the side bearer and a fuel plate having the under side ribbed to allow a free circulation of air.

The fuel plate is bolted to the web and carries the

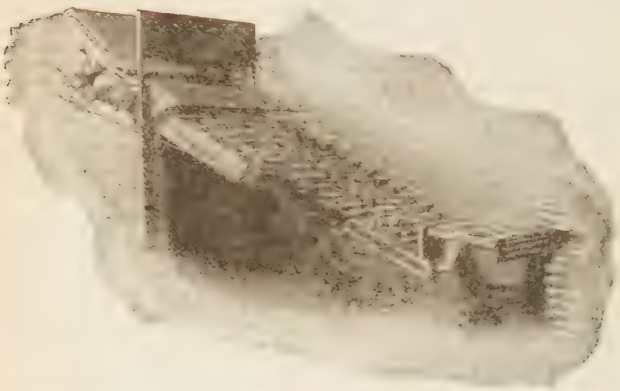


FIGURE 119.
SECTIONAL PERSPECTIVE OF THE ROney MECHANICAL STOKER.

fuel. The grates lie in a horizontal position across the furnace in the form of steps, and ample provision is made for the admission of air through the slotted webs. A fire brick arch is also sprung across the furnace, covering the upper portion of the grate.

This arch, being heated to a high temperature, serves in a measure to partly coke the coal as it passes under it. Air is also admitted on top of the coal at the front. This air is heated by its passage through a perforated tile over the dead plate and adjoining the

fire brick arch. Fig. 120 shows the location of the arch and tile.

In mechanical stokers of the under-feed type the air is supplied by forced draft.

The American stoker consists of a horizontal conveyor pipe into which the coal is fed from a hopper. The diameter of this pipe depends upon the quantity of coal to be burned, and varies from 4½ in. for the smaller sizes up to 9 and 10 in. for the larger sized stokers. The length of the conveyor pipe for the

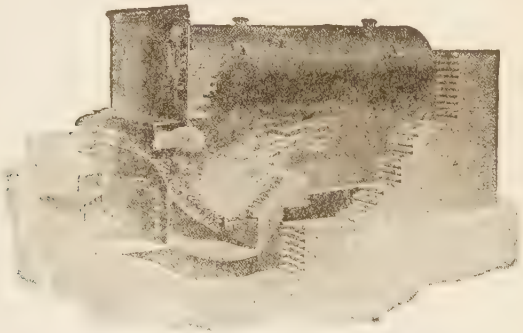


FIGURE 120.

standard 10-in. stoker is 72 in. Attached to the outer end of this conveyor pipe, and forming a part of it, is an iron box containing a reciprocating steam motor which, through the medium of a rocker arm and pawl and ratchet wheel, drives a screw conveyor shaft that slowly revolves within the pipe, thus forcing the coal forward and up through another box or trough, which latter is wholly within the furnace. Extending around the top edges of this box, and on a level with the grate bars, there is a series of tuyeres through which the air is forced.

These tuyeres, being at a high temperature, serve to heat the air in its passage through them, thus greatly

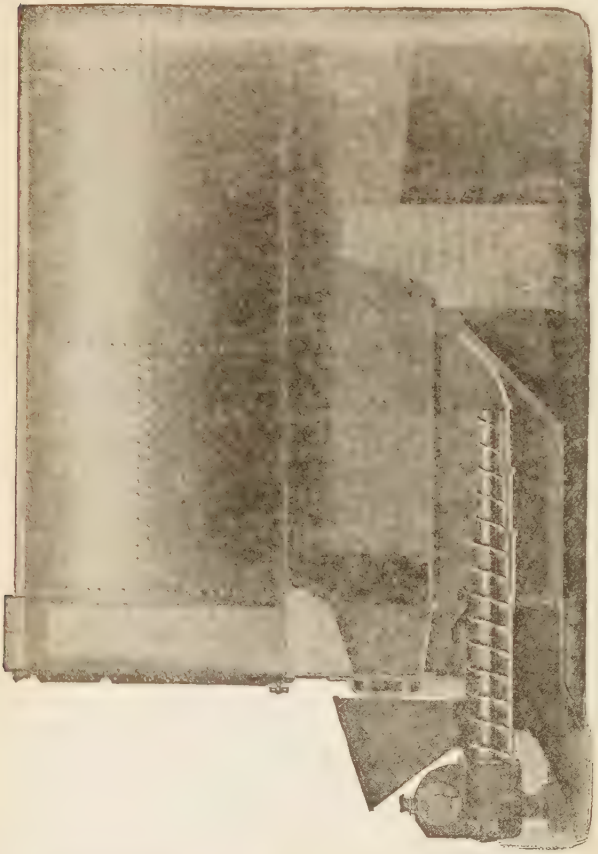


FIGURE 121.

TYPICAL SETTING OF AMERICAN STOKER UNDER A RETURN TUBULAR BOILER.

aiding combustion. Fig. 121 is a longitudinal sectional view of this stoker.

The speed of the screw conveyor is regulated by the hand throttle of the motor, according to the demand

for coal. With the 9-in. standard stoker from 350 to 1,200 lbs. of coal per hour may be burned. Fig. 122 is a view of the American stoker before being placed in position in the furnace.

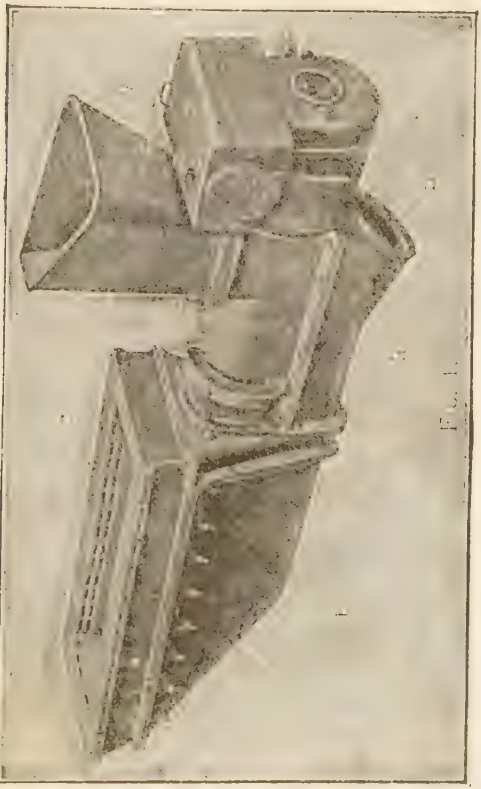


FIG. 122. STANDARD 8 AND 9 INCH AMERICAN STOKERS.

A. Coal hopper.
B. Conveyor pipe.
C. [illegible]es for introducing air to the fuel.
D. Opening to wind-box for air connections.
L. Wind-box for supplying air to tuyeres,
K. Automatic steam motor for driving conveyor.
R. Air pipe to wind-box.
S. Gas ducts for returning volatile products from entering coal to furnace.

The air jets, passing out from the tuyeres in a horizontal direction, and from opposite sides, cut through the rounded bed of coal and the gases are thus ignited and consumed immediately after being distilled

from the coal, while the pressure of the coal rising from underneath forces the already coked fuel over the edges of the trough or box onto the grates which occupy the space between the side walls and the coal trough. The air is first delivered from the fan into the air box that surrounds the coal trough on three sides and from thence it passes to the tuyeres. If this stoker is properly handled very good results may be obtained by its use, but, like all other devices for burning coal under boilers, it is bad policy to endeavor to force it beyond its capacity.

In the Jones under-feed stoker the coal is pushed forward and up into the furnace through a cast iron retort or trough. The impelling force is a steam ram connected to the outer end of the retort, and the speed of the ram is regulated automatically by the steam pressure, or by hand as desired. The coal is supplied to the ram through a cast iron hopper having a capacity of 125 to 140 lbs.

Forced draft is also employed in this stoker, the air being conducted from the fan or blower through galvanized iron pipes into the closed ash pit, which really forms an air box, as the space on either side of the retort that is usually occupied by grate bars is in this case covered by solid cast iron dead plates upon which the coked fuel lies until it is consumed. These plates, being hot, serve to heat the air coming in contact with them in its passage to the cast iron tuyeres through which it passes to the bed of burning fuel in the retort. Air entirely surrounds the retort on the sides and back end, and is at a constant pressure in the ash pit, but can only pass into the furnace through the tuyeres, the jets of air cutting through the rounded heap of incandescent fuel from opposite sides and in a direction inclined upwards.

Coal is supplied to the hopper either by hand, or by mechanical means where the plant is fitted with coal-handling machinery. The opening through which it passes from the hopper to a position in front of the ram is 8 × 10 in. in size. Each charge of the steam

FIGURE 123.

ram carries forward 15 to 20 lbs. of coal. Connected to the ram and moving in conjunction with it is a long rod extending through the retort near the bottom. Upon this rod are carried shoes that act as auxiliary plungers and facilitate the movement of the coal.

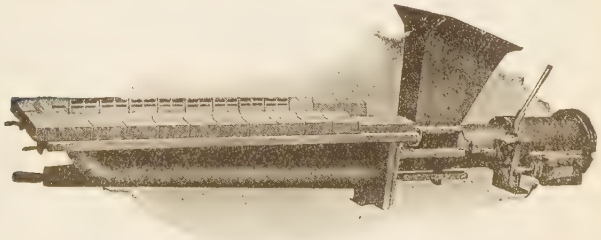


FIGURE 124.

Fig. 123 is a sectional view of the Jones stoker, showing the machine full of coal, with the ram ready to make a charge. Fig. 124 shows the stoker complete before being placed in the furnace.

It is claimed by the builders of under-feed stokers, and the claim appears to have good foundation, that by pushing the green coal up so as to meet the upper crust of glowing fuel the gases on being distilled immediately come in contact with and are consumed by the burning mass, and the formation of smoke is thus prevented. Both the Jones under-feed and the American stokers have proved to be very successful in the burning of the cheaper bituminous coals of the West. One feature tending to commend them is the fact that practically all of the coal is utilized, there being no waste caused by the slack coal or fine screenings dropping through the grate bars into the ash pit unconsumed.

A good substitute for the mechanical stoker is an outside furnace, by which is meant a boiler installation having the furnace in front of instead of underneath the boiler. One of the principal hindrances to good combustion in the ordinary type of boiler furnace is the fact that the temperature of the boiler shell or water tubes with which the gaseous products of combustion come in contact can never be higher than the temperature of the water contained within the boiler. This temperature ranges from 297° for steam at 50 lbs. gauge pressure, up to 407° for 255 lbs. pressure, while the temperature of the furnace, according to Dr. Thurston and other high authorities, ranges from 2,010° to 2,550°.

It is evident that perfect combustion does not take place until these high temperatures are reached. Each time the furnace is charged with fresh coal, especially if the boiler be hand-fired, a large volume of volatile gases is liberated but not consumed. If these gases are allowed to immediately come in contact with a

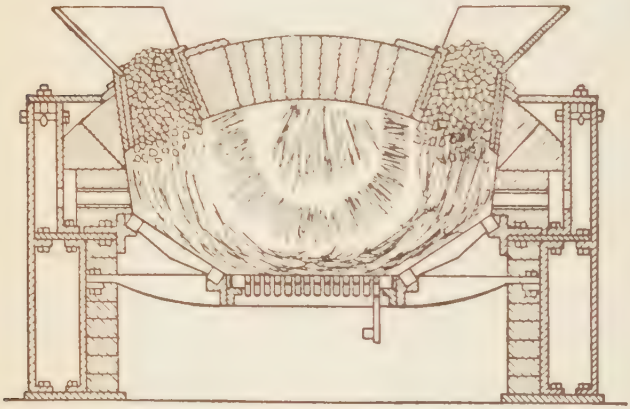
comparatively cool surface, as for instance the heating surface of the boiler, the result is a cooling of the gases, incomplete combustion and the formation of smoke and soot. If on the other hand the furnace is so constructed that these gaseous products first impinge against hot surfaces, such as fire brick arches or bafflers that have a temperature corresponding to that of the furnace, good combustion is assured. This condition is in a large degree attained by the use of outside furnaces that permit the construction of a fire brick arch to cover the entire grate surface.

Front View.
FIGURE 125.

The Burke furnace, patented by James V. Burke of Chicago, is a notable example of this type of furnace. It is applicable to any type of stationary boiler. Fig. 125 shows this furnace as applied to tubular boilers. It consists of a fire brick arch extending from 6 to 8 ft. outwards from the boiler front and of a width to correspond to the diameter of the boiler. The arch rests securely upon brick work inclosed in a well ventilated iron casing. There is practically no heat radiated from this furnace, all the heat generated

by it passing to the boiler. The central portion of the grate bars consists of shaking grates, while the side bars are stationary and inclined.

Fig. 126 is a sectional view and will serve to illustrate the construction of this furnace. The coal is fed through pockets on top on each side of the arch, the larger furnaces having two pockets on each side and the smaller sizes one. The doors in front are only



CROSS SECTION.
FIGURE 126.

opened for the purpose of cleaning fires or when first starting fires. The air is supplied by way of the ash pit, passing up through the grate bars. A portion of the air supply is also drawn through the ventilators and passes to the upper part of the furnace. The arch extends under the front end of the boiler 6 or 8 in., and there is a bridge wall about 4 ft. back from the front against which the gases from the furnace impinge.

There are 42 sq. ft. of grate surface in the larger

sizes, and 22 sq. ft. in the smaller size. Good combustion is attained in this furnace, owing to the fact that the gases as they are distilled from the coal come immediately in contact with the highly heated surface of the arch directly over the fire.

ADDENDA

Types of Modern Water Tube Boilers Described and Illustrated

The numbers designating the cuts are herein duplicated, beginning with Figure 5

THE HEINE SAFETY BOILER.

Figure 5 shows a general view of the Heine water tube boiler.

The boiler is composed of the best lap welded wrought iron tubes, extending between and connecting the inside faces of two "water legs," which form the end connections between these tubes and a combined steam and water drum or "shell," placed above and parallel with them. (Boilers over 200 horsepower have two such shells.) These end chambers are of approximately rectangular shape, drawn in at top to fit the curvature of the shells. Each is composed of a *head plate* and a *tube sheet*, flanged all around and joined at bottom and sides by a butt strap of same material, strongly riveted to both. The water legs are further stayed by *hollow stay bolts* of hydraulic tubing, of large diameter, so placed that two stays support each tube, and hand hole and are subjected to only very slight strain. Being made of heavy metal, they form the strongest parts of the boiler and its natural supports. The WATER LEGS are joined to the shell by *flanged and riveted joints* and the drum is cut away at these two points to make connection with inside of water leg, the opening thus made being strengthened by bridges and special stays, so as to preserve the original strength.

THE SHELLS are cylinders with *heads dished* to form

parts of a *true sphere*. The sphere is everywhere as strong as the circle seam of the cylinder which is well known to be twice as strong as its side seam. Therefore these heads require no stays. Both the cylinder and its spherical

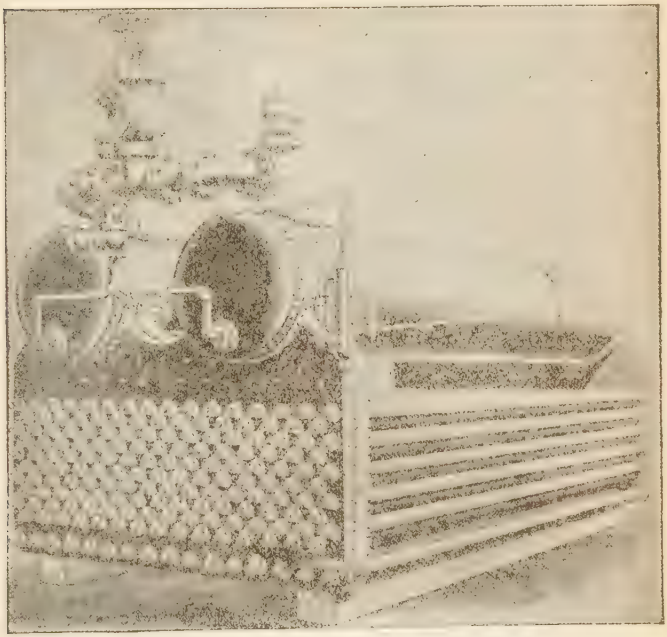


FIG. 5
375 H. P. HEINE BOILER

heads are therefore *free to follow* their *natural lines of expansion* when put under pressure. Where flat heads have to be braced to the sides of the shell, both suffer local distortions where the feet of the braces are riveted to them, making the calculations of their strength fallacious. To

the bottom of the front head a flange is riveted into which the feed pipe is screwed. This pipe is shown in the cut with *angle valve* and *check valve* attached.

On top of shell near the front end is riveted a *steam nozzle* or saddle, to which is bolted a tee. This tee carries the *steam valve* on its branch, which is made to look either to front, rear, right or left; on its top the *safety valve* is placed. The saddle has an area equal to that of stop valve and safety valve combined. The rear head carries a *blow-off flange* of about the same size as the feed flange, and a *manhead* curved to fit the head, the manhole supported by a strengthening ring outside. On each side of the shell a square bar, the *tile-bar*, rests loosely in flat hooks riveted to the shell. This bar supports the *side tiles* whose other ends rest on the *side walls*, thus closing in the furnace or flue on top. The top of the tile bar is two inches below *low water line*. The bars rise from front to rear at the rate of one inch in twelve. When the boiler is set, they must be exactly level, the whole boiler being then on an incline, i. e., with a fall of one inch in twelve from front to rear.

It will be noted that this makes the height of the *steam space* in front about *two-thirds* the diameter of the shell, while at the rear the *water* occupies *two-thirds* of the shell, the whole contents of the drum being equally divided between steam and water.

The tubes extend through the tube sheets into which they are expanded with roller expanders; opposite the end of each and in the head plates is placed a hand hole of slightly larger diameter than the tube, and through which it may be withdrawn. These hand holes are closed by small cast iron hand hole plates, which can be easily removed for the purpose of cleaning or inspecting a tube. Figure

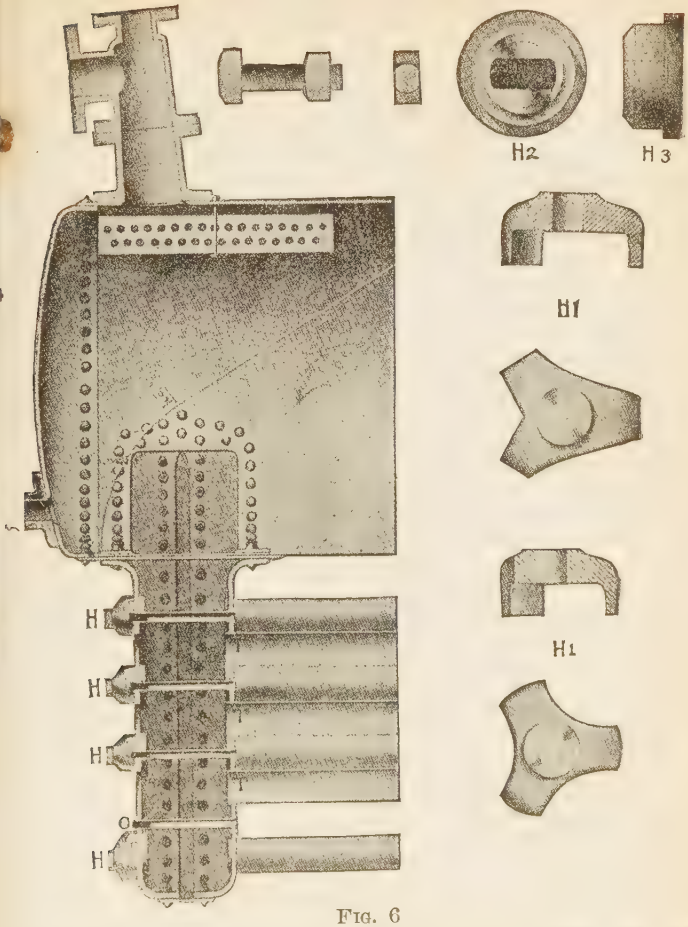


FIG. 6

DETAIL OF WATER-LEG, HAND HOLE PLATES AND YOKES, ETC., OF HEINE BOILERS

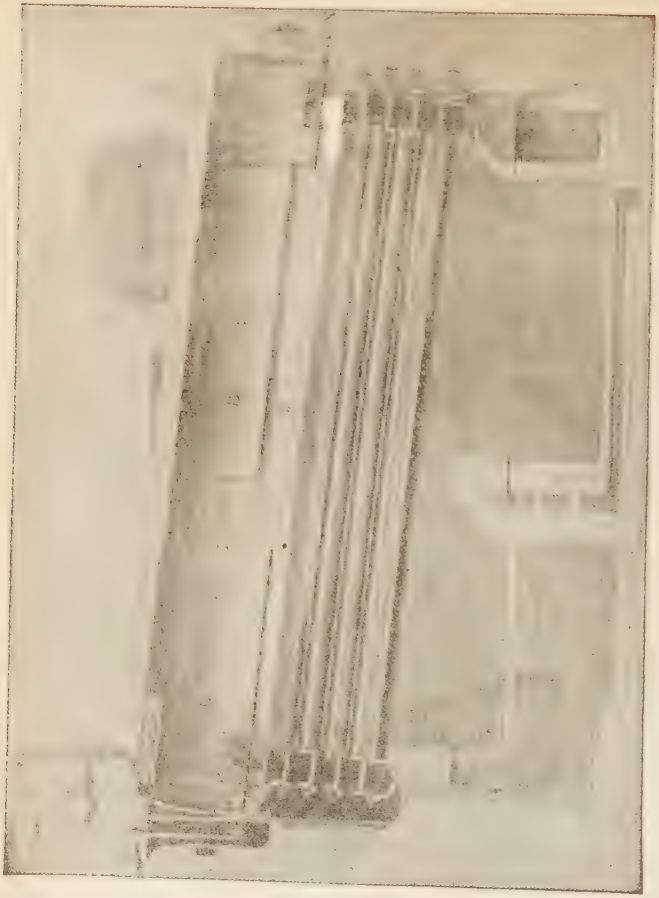


FIG. 7
SIDE SECTIONAL VIEW OF HEINE BOILER

6 shows these hand hole plates, marked H. In the upper corner one is shown in detail, H^2 being the top view, and

H^3 a side view, while H^1 is the yoke or crab placed outside to support the bolt and nut. Figure 7 is a longitudinal section showing inside construction. The mud drum D is located well below the water line, parallel to and three inches above the bottom of the shell. It is of oval section, slightly smaller than the manhole. It is entirely enclosed, except about eighteen inches of its upper portion at the front end, which is cut away nearly parallel with the water line. The mud drum D is made of strong sheet iron, with cast iron heads, and its action is as follows:

The feed water enters it through the pipe F about one-half inch above its bottom; even if it has previously passed the best heaters it is colder than the water in the boiler. Hence it drops to the bottom, and, impelled by the pump or injector, passes at a *greatly reduced speed* to the rear of the mud drum. As it is gradually heated to near boiler temperature it rises and flows slowly in reverse direction to the open front of the mud drum; here it passes over in *a thin sheet* and is immediately swept backward into the main body of water by the swift circulation, thus becoming *thoroughly mixed* with it before it reaches the tubes. During this process the mud, lime, salts, and other precipitates are deposited as a sort of semi-fluid "sludge" near the rear end of the mud drum, whence it is blown off at frequent intervals through the blow-off valve N. As the speed in the mud drum is only about one-fiftieth of that in the feed water pipe, plenty of time is given for this action. Any precipitates which may escape the mud drum at first, will of course form a scale on the inside of the tubes, etc. But the action of expansion and contraction cracks off scale on the *inside* of a tube *much faster* than on the *outside*, and then the circulation sweeps the small chips, like broken

egg-shells, upward, and as they pass over the mouth of the mud drum *they drop in the eddy*, lose velocity in this slow current and fall to the bottom, and, being pushed by the feed current to the rear end, are blown off from the mud drum with other refuse. On opening a Heine boiler after some months' service, such bits of scale, whose shape identifies them, are always found in the mud of the mud drum. *Very little loose scale* is found on the bottom of the water legs; the current through the lower tubes, always the swiftest, brushes too near the bottom to allow much to lodge there.

This explanation of the action of the mud drum shows how *the inside of the tubes may be kept clean*. To keep the *outside clear of soot and ashes* which deposit on, and sometimes even bake fast to the tubes, each boiler is provided with two special nozzles, with both side and front outlets, a short one for the rear, a long one for the front. They are of three-eighth inch gas pipe and each is supplied with steam by a one-half inch steam hose. The nozzle is passed through each stay bolt in turn, and thus delivers its side jets on the three or four tubes adjacent, with the full force of the steam, at the short range of two inches, *knocking the soot and ashes* off completely, while the end jet carries them into the main draft current, to lodge at points in breeching or chimney base convenient for their ultimate removal. An inspection of the cuts will show that the stay bolts are so located that the nozzle can in turn be brought to bear on all sides of the tubes. As soon as the nozzle is withdrawn from the stay bolt this is closed air-tight by a plain wooden plug.

In cleaning a boiler it is only necessary to *remove every fourth or fifth handhole plate* in the front water leg; the

water hose, supplied with a short nozzle, can be entered in all the adjacent tubes, owing to the ample dimensions of the water leg. In the rear water leg only one or two hand-holes in the lower row need be opened to let the water and debris escape. The others in rear water leg are frequently *left untouched for years.* A lamp or candle hung on a wire through the manhead may be held opposite each tube so that it can be perfectly inspected from the front.

The feed pipe F enters the mud drum through a loose joint in front, and the blow-off pipe N is screwed tightly into its rear head, passing by a steam-tight joint through the rear head of the shell. Just under the steam nozzle is placed a dry pipe A. L is a deflection plate, extending from the front of the shell, and inclined upwards, beyond the mouth of the front water leg. The throat or mouth of each water leg is large, to equal in area the total tube area, and where it joins the shell it increases gradually in width by double the radius of the flange. In the setting, the front water leg is placed firmly on a set of strong cast iron columns, bolted and braced together by the door frames, dead plate, etc., forming the fire front. This is the fixed end. The rear water leg rests on rollers, free to move on cast iron plates set in the lower masonry of the rear wall. The brick work does not close in entirely to the boiler, the space between being filled with tow or waste saturated with fire clay or other refractory, but pliable material, thus leaving the boiler and its walls each free to move separately during expansion or contraction, without disturbing any joints in the masonry. On the lower, and between the upper tubes, are placed light fire brick tiles. The lower tier extends from the front water leg to within a few feet of the rear one, leaving there an upward

passage across the rear ends of the tubes for the flame and gases. The upper tier closes into the rear water leg, and extends forward to within a few feet of the front one, thus leaving an opening for the gases in front. The side tiles extend from the side walls to tile bars, and close up to the front water leg, and front wall, and leave open the final uptake for the waste gases over the back part of the shell, which is here covered above the water line with a row of lock fire brick, resting on the tile bars. The rear wall, and one parallel to it, are arched over the shell a few feet forward, and form the uptakes.

THE BABCOCK & WILCOX WATER TUBE BOILER.

Description. Figure 8 presents a side view of the Babcock & Wilcox boiler, and Figure 9 a partial section.

This boiler is composed of lap-welded wrought iron tubes, placed in an inclined position and connected with each other, and with a horizontal steam and water drum, by vertical passages at each end, while a mud-drum is connected to the rear and lowest point in the boiler.

The end connections are in one piece for each vertical row of tubes, and are of such form that the tubes are "staggered" (or so placed that each row comes over the spaces in the previous row). The holes are accurately sized, made tapering, and the tubes fixed therein by an expander. The sections thus formed are connected with the drum, and with the mud-drum also by short tubes expanded into bored holes, doing away with all bolts, and leaving a clear passageway between the several parts. The openings for cleaning opposite the end of each tube are closed by hand-hole plates, the joints of which are made in the most thorough manner, by milling the surfaces to accurate metallic

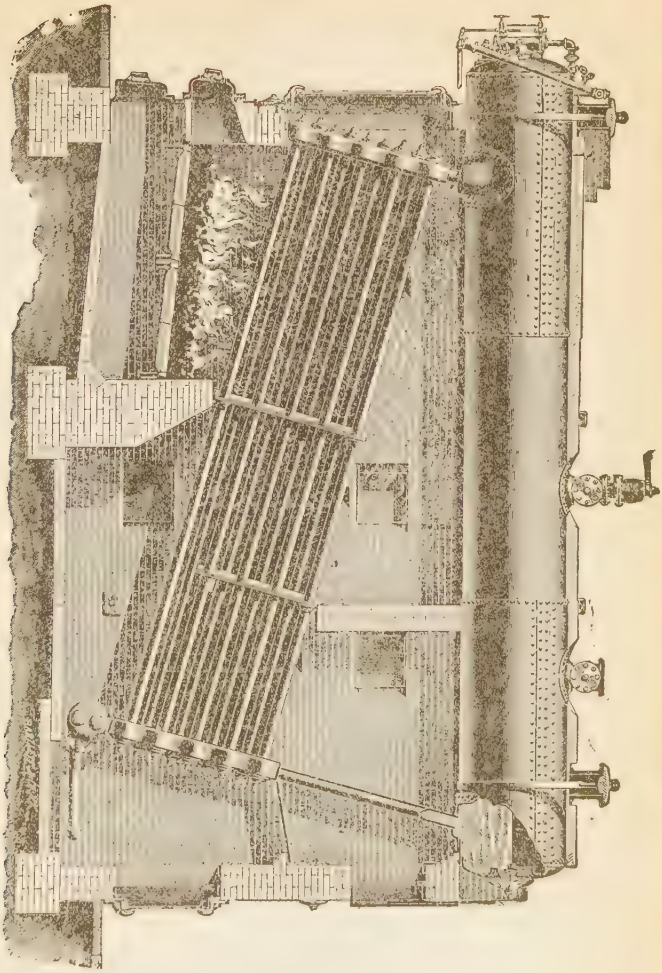


FIG. 8

SIDE VIEW OF BABCOCK & WILCOX BOILER OF WROUGHT STEEL CONSTRUCTION

contact, and are held in place by wrought-iron forged clamps and bolts. They are tested and made tight under a hydrostatic pressure of 300 pounds per square inch, *iron to iron, and without rubber packing or other perishable substances.*

The steam and water drums are made of flange iron or steel, of extra thickness, and double riveted. They can be made for any desired pressure, and are always tested at 50 per cent above the pressure for which they are constructed. The mud-drums are of cast iron, as the best material to withstand corrosion, and are provided with ample means for cleaning.

Erection. In erecting this boiler, it is suspended entirely independent of the brickwork, from wrought iron girders resting on iron columns. This avoids any straining of the boiler from unequal expansion between it and its enclosing walls, and permits the brickwork to be repaired or removed, if necessary, without in any way disturbing the boiler. All the fixtures are extra heavy and of neat designs.

Operation. The fire is made under the front and higher end of the tubes, and the products of the combustion pass up between the tubes into a combustion chamber under the steam and water drum; from thence they pass down between the tubes, then once more up through the spaces between the tubes, and off to the chimney. The water inside the tubes, as it is heated, tends to rise towards the higher end, and as it is converted into steam—the mingled column of steam and water being of less specific gravity than the solid water at the back end of the boiler—rises through the vertical passages into the drum above the tubes, where the steam separates from the water and the latter flows back to the rear and down again through the tubes in a con-

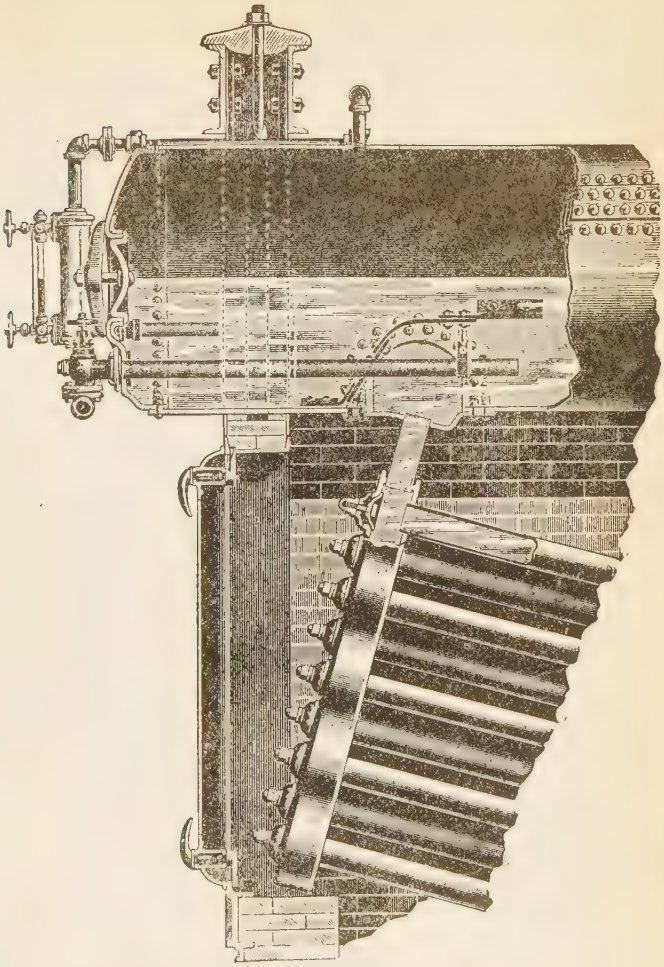


FIG. 9

PARTIAL VERTICAL SECTION BABCOCK & WILCOX BOILER

tinuous circulation. As the passages are all large and free, this circulation is very rapid, sweeping away the steam as fast as formed, and supplying its place with wa-

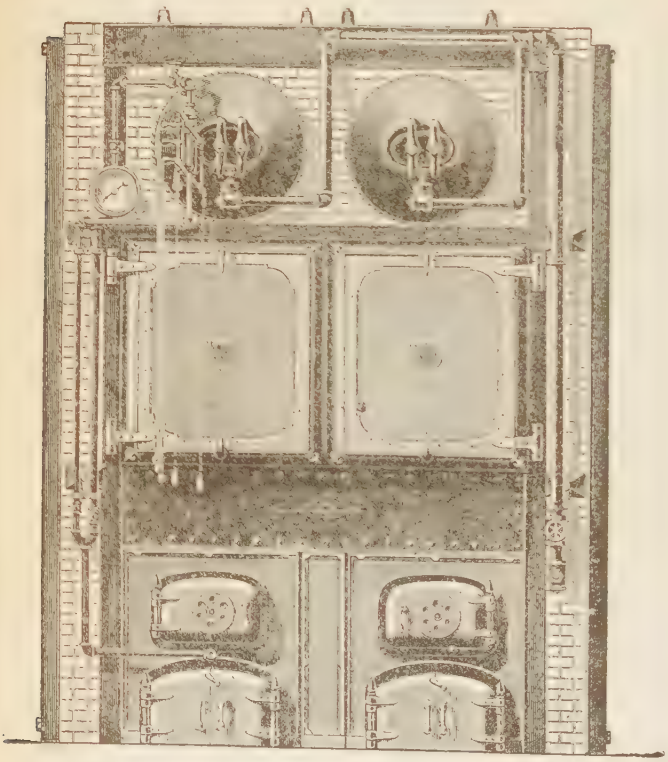


FIG. 10
STANDARD FRONT OF BABCOCK & WILCOX BOILER

ter; absorbing the heat of the fire to the best advantage; causing a thorough commingling of the water throughout the boiler and a consequent equal temperature, and pre-

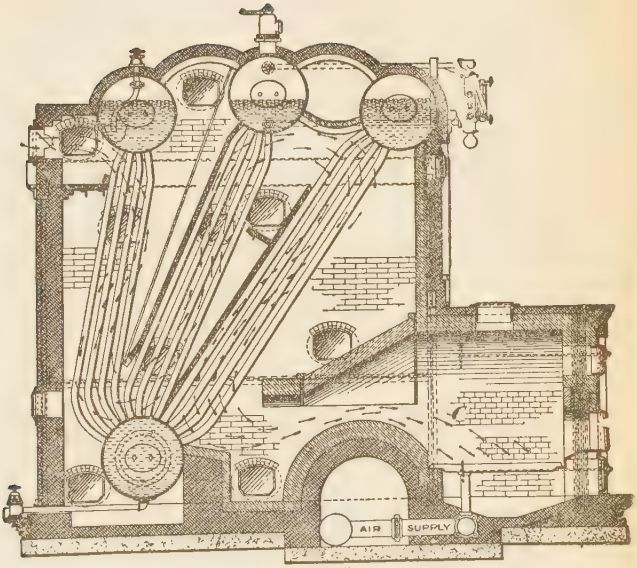


FIG. 11
SECTIONAL SIDE ELEVATION OF THE STIRLING BOILER AND BAGASSE FURNACE.

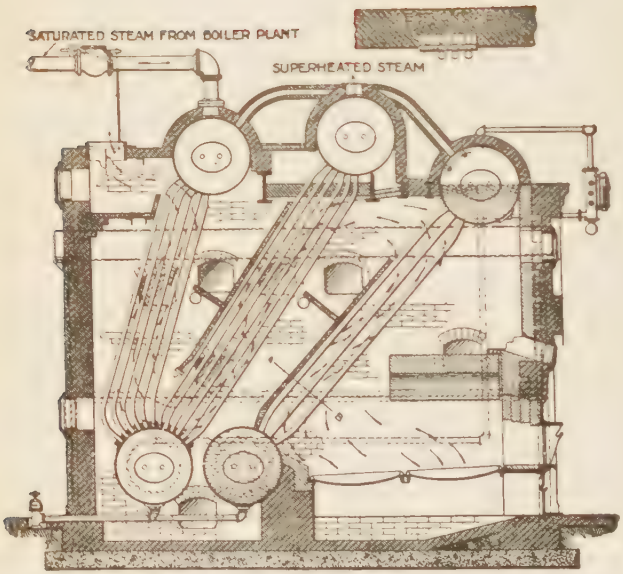


FIG. 12

SECTIONAL SIDE ELEVATION OF THE STIRLING INDEPENDENTLY FIRED SUPERHEATER.

venting, to a great degree, the formation of deposits or incrustations upon the heating surfaces, sweeping them away and depositing them in the mud-drum, whence they are blown out.

The steam is taken out at the top of the steam-drum near the back end of the boiler after it has thoroughly separated from the water, and to insure dry steam, a perforated dry-pipe is connected to the nozzle inside the drum.

THE STIRLING WATER-TUBE SAFETY BOILER.

The Stirling boiler, Figures 11 and 12, consists of three upper or steam drums, each connected by a number of tubes (called a "bank") to a lower or mud drum. Suitably disposed fire tile baffles between the banks direct the gases into their proper course. Shorter tubes connect the steam spaces of all upper drums, also water spaces of front, and middle drums. The boiler is supported on a structural steel framework, around which is built a brick setting, whose only office is to provide furnace space, and serve as a housing to confine the heat. The entire front is of metal of appropriate and artistic design. These parts, together with the usual valves and fittings, constitute the completed boiler, which represents the acme of simplicity and eliminates the complication of the older types.

The drums vary from 36 to 54 inches in diameter and are made of the best open hearth flange steel. The plates extend the entire distance between heads, hence there are no *circular seams*. The longitudinal seams—which are double or triple riveted according to the working pressure to be carried—are so placed that they are not exposed to high temperature. The drum heads are hydraulically dished to proper radius; each drum is provided with one

manhole, and the manhole plate and arch bars are of wrought steel; four manhole plates, which can be removed in ten minutes, give access to the entire interior of the boiler, and expose every tube end, rivet and joint. The drum interiors are perfectly clear; there are no baffles, stays, tie-rods, mud pipes or other obstructions in them.

The tubes are best lap-welded mild steel. They are slightly curved at the ends to permit them to enter the drums normally and to provide for free expansion of the boiler when at work. The tubes are expanded directly into reamed holes in tube sheets of the drums, hence the annular recess between tubes and the cast headers of some types of boiler is eliminated, and failure of tubes by pitting through corrosion, caused by accumulation of soot in these recesses, is avoided. There are no short nipples and no tube joints in places which can be reached only by jointed handles on the tube expander, rendering it impossible to determine when the tube has been properly expanded. In the Stirling boiler every tube end is visible and accessible. As the entire weight of boiler and contents is supported on the steel frame work, cracking of the setting, due to unequal settlements, is obviated, and no blocking is needed when the brick work has to be repaired. The design of frame work can be modified to suit special conditions. The brick setting is so clearly shown in Figures 11 and 12 that an extended description is unnecessary. No special shapes or other material not found in open market are needed. Any necessary repairs to the brick work can be made without disturbing the boiler connections. In the design of the Stirling furnace it will be seen by reference to Figures 11 and 12, that a fire brick arch is sprung over the grates, and immediately in front of the first bank of tubes. The

large triangular space between boiler front, tubes and mud drum is available for a combustion chamber, and for installation of sufficient grate surface to meet the requirements of the lowest grades of fuel.

Baffles and Course of Gases. The baffle walls rest directly upon the tubes, and guide the course of the gases up the front bank, down the middle and up the rear bank, thus bringing them into such intimate contact with the

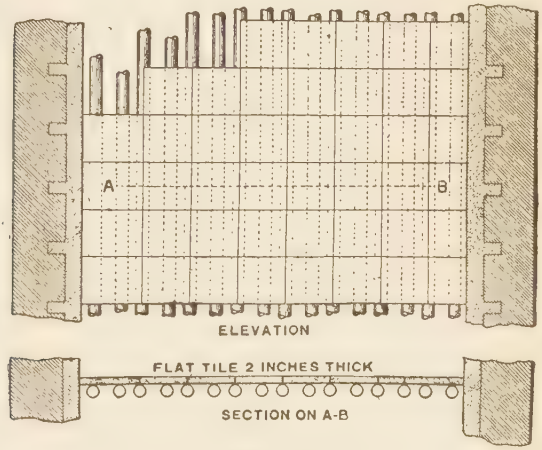


FIG. 13

ELEVATION AND SECTION OF FIRETILE BAFFLES IN STIRLING BOILERS

boiler surface that the heat is quickly and thoroughly extracted from them. In no other boiler are the gases compelled to travel as far before reaching the stack, and the effect upon economy is evident. The baffles are made of plain rectangular firetile carried in stock by all fire-brick dealers, in contrast to the special formed bricks (obtainable only from the manufacturer) required by many types

of water-tube boiler. Another marked advantage of the Stirling baffles is that since no tubes pass *between* or through the tiles (see Fig. 130), they are not pried apart and made leaky by distorted tubes: they can be removed and replaced without disturbing a tube. Baffles built across the tubes, as in many boilers, are damaged by pulling a faulty tube through them, and can be repaired in but one way—by removal of every tube necessary to permit a man to crawl in and reach the defective spot.

Simplicity. There are no details of complicated shape: no flat surfaces, tie-rods, water-legs, headers, return-bends, outside circulating pipes to plug up: no multitudinous handhole plates to be removed and packed with gaskets, or to be ground and scraped to a fit whenever boiler is opened: no baffles or mud pipes in the drums: no short nipples, seams exposed to heat, or parts inaccessible for cleaning.

Expansion and Contraction. In the Stirling, the mud drum is not embedded in brickwork, but is suspended on the tubes which connect it with the upper drums.

In consequence of this construction, not only may the mud drum with perfect freedom move an amount representing the resultant expansion of the boiler, but any difference in expansion between the individual tubes, such as caused when one side of the furnace is being cleaned and other side is excessively hot, is taken up by the curve in the tube. The boiler therefore stays tight, and is entirely free from the stresses and frequent leaks caused by unequal expansion of straight tubes rigidly connected at each end to headers, water-legs, or large drums. It will thus be seen that the bent tube performs in the boiler the same function as an expansion loop in a steam line.

Rapid Circulation. The path of the circulation in the Stirling is as follows: The water is fed into upper rear drum, passes down the rear bank of tubes to the lower drum, thence up the front bank to forward steam drum. Here the steam formed during passage up the front bank disengages and passes through the upper row of cross tubes into the middle drum, while the *solid water* passes through the lower cross tubes into middle drum, then down the middle bank to the lower drum, from which it is again drawn up the front bank to retrace its former course until it is finally evaporated. The steam generated in the rear bank passes through cross tubes to the center drum.

The temperature of gases in contact with the tubes will evidently be greatest at the bottom of the front bank, and gradually decreases as the gases proceed along their course to the breeching. Obviously then the velocity of water circulation and quantity of steam generated will be a maximum in the front bank; in the rear bank there is a slow circulation downward equal to the quantity of water evaporated in the other two banks.

Rapid circulation is essential for the following reasons:

(1) To keep all parts of the boiler at practically the same temperature, thus eliminating severe stresses due to unequal expansion.

(2) To permit quick raising of steam and rapid response to sudden demands on the boiler capacity.

(3) To sweep away from the heating surfaces all steam bubbles as fast as formed, and thereby prevent "steam pockets," which quickly burn out the tubes. This is so particularly the case where intense local heating occurs due to use of gas or oil fuel, that some types of boiler fairly well adapted to coal cannot be successfully used with these fuels.

The third requirement is met only indifferently or, not at all in those types of boilers in which tubes often numbering as many as eighteen must discharge their entire content of steam and water through a narrow water-leg, or, worse still, through a single nipple whose cross section is equal to that of but one tube. At 150 pounds' gauge one cubic foot of water, when converted into steam, will have a volume of about 151 cubic feet. In consequence of this great increase in volume, as soon as the boiler is forced, the nipple area becomes insufficient, steam pockets form in the lower tubes, which then become overheated, and buckle and leak, and finally burn out. So inadequate are these nipples and headers that recent experiments of M. Brull have shown that in boilers whose circulation is constricted by nipples or narrow water-legs, the circulation in the upper tubes *reverses*, that is, it goes from the front to rear instead of in the opposite way, as intended. In consequence of this, much matter suspended in the water is swept into the bottom tubes, which fact, in connection with the steam pockets, explains why those tubes so rapidly fail.

In the Stirling boiler there is *no constriction of the circulation*, as each tube discharges directly into the drums, without intervention of headers, nipples or water-legs. The nearly vertical position of the tubes also promotes rapid circulation, hence steam pockets cannot form, and a fruitful cause of interrupted service and tube renewals is thus eliminated.

A most fruitful cause of burnt tubes is a piece of scale which becomes detached and falls on the bottom of the tube, and the spot under it is certain to burn out quickly. The Stirling is free from this source of tube destruction, because while the scale will not form in the hotter tubes

unless the boiler is neglected, even if it does form owing to such neglect and a piece becomes detached, it will slide down to the mud drum instead of lodging.

Cleaning the Interior. By removing four manhole plates, which can be done in ten minutes, the entire boiler interior is accessible for cleaning. From the preceding discussion it is evident that the precipitates are settled into the mud drum, whence they are blown off at intervals; the scale is practically confined to the rear bank of tubes, and by reason of escaping the high temperatures it is soft and easily detached. Consequently it happens in most cases that only the rear bank needs cleaning each time the boiler is opened, while the others need only occasional attention. The scale is quickly and cheaply removed by a "turbine cleaner," described and illustrated in the section on Boiler Operation.

Cleaning the Exterior. Ample cleaning doors are provided both in the sides and rear of the setting, so that the exterior of the heating surfaces may be kept clean and all accumulations of soot, ashes, etc., blown off as rapidly as they form, by using a steam blower-pipe which is furnished with every boiler.

Durability. By reason of the elimination of thick plates and riveted joints exposed to the fire; cast metal of all kinds; parts of irregular shape and uncertain strength; stresses due to unequal expansion, multitudes of caps, joints and nipples, and similar objectionable details, the Stirling boiler is free from parts liable to get out of order. The prevention of scale deposits in the hottest tubes; the perfect facilities for keeping the boiler clean; the rapidity of water circulation and impossibility of forming steam pockets, all combine to protect the tubes against burning

out. Hence the necessity of repairs to the boiler itself is extremely remote. The setting is simple and substantial and not subject to derangement other than the natural wear of surface lining.

MAXIM WATER-TUBE BOILER.

The Maxim water-tube boiler (Figs. 14 and 15) consists of two drums, one above the other, connected by tubes.

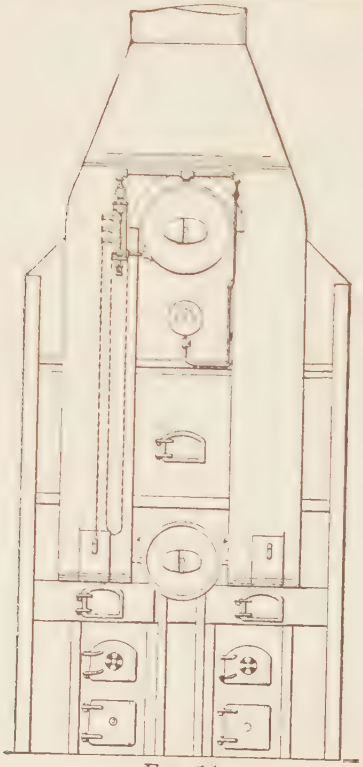


FIG. 14

FRONT ELEVATION MAXIM BOILER

Each tube has two bends, thus providing for unequal expansion and contraction. The space between the tubes is greater than the diameter of the tubes, so that any tube can be passed through the space between. The tubes, which are nearly vertical, are arranged in parallel rows, the lower

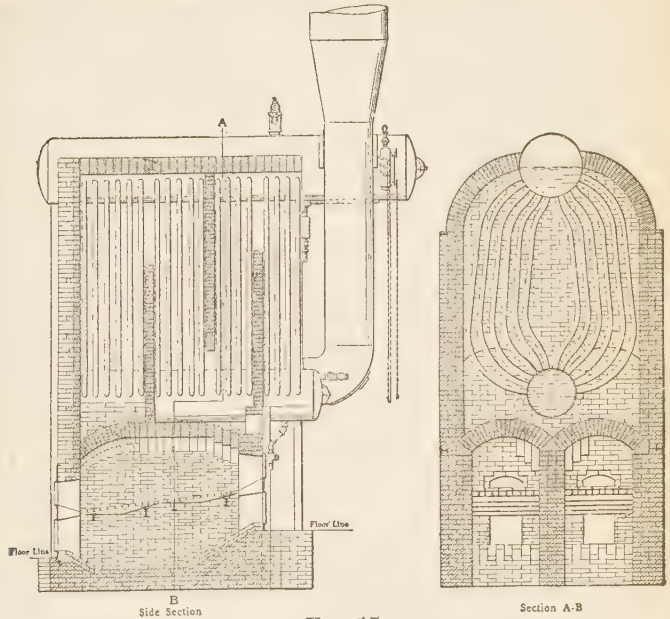


FIG. 15
MAXIM BOILER

tube plates being cylindrical, and the bottom end of every tube is opened to the lower drum, thus enabling the flue dust, mud and loose scale to drop away from the heating surface. There are no riveted joints exposed to the flames or fire gases, and there are no cylinders subject to external pressure.

The heating surface is arranged so as to break up the current of heated gases, which travels three times the length of the tube, the form and direction of the gas current being changed seven times between the furnace and chimney, to insure a low temperature to the escaping gases.

The circulation is constant, the steam generation being provided for by a third set of tubes, which are subjected to the heat of the gases just before reaching the chimney. As the cold feed water meets the gases as they leave the boiler, the front section of the tube, which receives the feed water, acts as an economizer and purifier.

The furnace of this type of boiler is constructed of fire-brick, and is built under the boiler. Between the furnace and each combustion chamber there is a throat contracted to the proper size for drafts, the purpose being to insure a better combustion by an intimate mixture of gases. The illustrations show a side view, sectional view through A B, and a front view.

THE BIGELOW-HORNSBY WATER-TUBE BOILER.

The American rights to manufacture the Hornsby water-tube boiler, which has been on the English market five years, were recently acquired by the Bigelow Company, of New Haven, Conn., hence in this country it will be known as the "Bigelow-Hornsby" boiler. It has been re-designed in a measure, to meet the requirements of American high-pressure practice. A general idea of its construction may be obtained from Fig. 16, which shows a section through the setting. It will be noted that the boiler is supported entirely from the overhead beams, leaving the lower part free to respond to expansion stresses. The front tubes are inclined at an angle of 68 degrees, and the rear units

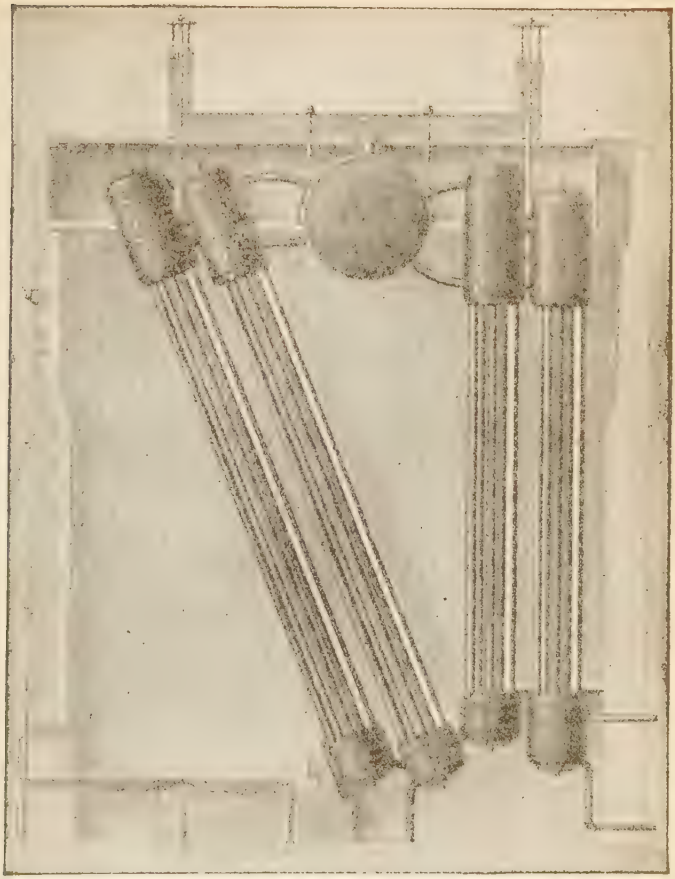


FIG. 16

SECTION THROUGH SETTING OF BIGELOW-HORNSBY BOILER

are vertical, as shown. Only two lengths of tubes are used. Each section is independent of its neighbor, except the nipples connecting with the steam drum and the equal-

izing nipples which connect the bottom drums of the rear sections. This flexible form of construction permits the building of very large units, even larger than the 2,200-horsepower Hornsby boilers at the Bow Street station of the Charing Cross & London Electricity Works, where four boilers containing 21,700 square feet of heating surface evaporate in regular service 110,000 pounds of water per hour. A large percentage of the heating surface is exposed to the radiant heat of the furnace, and to the first pass of gases, before these have reached any other heating surface. The tubes of the front unit, which are located in front of the baffles, comprise more than 12 per cent of the heating surface of the boiler.

The feed-water is admitted into the bottom drum of the rear unit and is advanced gradually from the coldest to the hottest portion of the boiler, first passing up the entire length of the tubes in the rear unit, then from the top of those tubes to the unit immediately in front, down this unit, and up the two front units and back through the steam drum to the first vertical unit at the rear of the steam drum. There is also, as may be conceived, a rapid local circulation in each of the units while the general circulation is going forward. The speed of the feed-water up the rear unit being regulated by the amount of steam generated, ample time is permitted for sediment and scale-forming matter to be deposited in the bottom drum of this unit. All liberation of steam from the water surfaces takes place in the upper drums and is entirely unrestrained, the full area of the tube openings communicating with the drums. The transfer of steam and water between units occurs through separate nipples, and the water nipples are required merely to care for the general circulation through the boiler.

The arrangement of this boiler is such that it can be baffled so that the products of combustion are carried uniformly over the heating surfaces in thin layers, there being no large unrestricted paths parallel to the heating surfaces through which the gases can flow in their passage to the

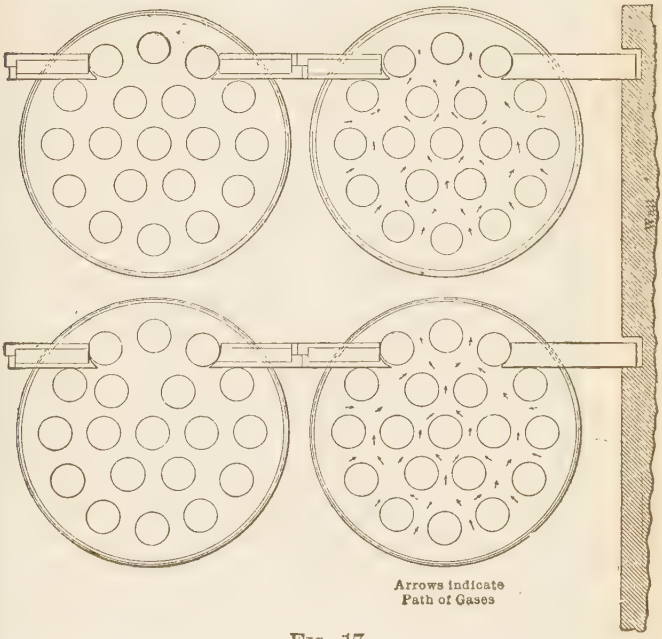


FIG. 17

uptake. Fig. 17, which is a horizontal section through some of the rear units, shows how they are arranged with reference to the gas currents, and how the baffle-plates serve to guide the gases through in substantially uniform passages.

The smoke flue can be taken off at the back of the boiler at any point between the top and bottom. All the tubes are straight and every tube in each section can be reached for cleaning by the removal of a single manhole cover.

The boiler is built to a factor of safety of five for 200 pounds working pressure, and it is stated that a test section has been subjected to hydrostatic pressure of 1,000 pounds without rupture. The ratio of grate surface to heating can be made as low as 1 to 26.

WICKES VERTICAL WATER-TUBE BOILER.

This boiler (see Fig. 18) consists primarily of two cylinders joined together by straight tubes, which are divided by a fire brick tile passing through their center into two compartments. The whole is then erected in a vertical position and surrounded by brickwork.

Drum. The two cylinders are duplicates in their diameter and general construction, but differ in height and arrangement of convexed heads. The top cylinder, designated hereafter, from its use, as the steam drum, is the longer, the length and diameter being varied in accordance with the size of boiler desired and local requirements.

The bottom cylinder, designated hereafter, from its use, as the mud drum, is the shorter, and is varied in the dimension as to diameter and length in accordance with the power of boiler required and local conditions. Both drums are closed at one end with the tube sheet, and at the other end with convexed heads.

Tubes. The mud and steam drums are joined together by the tubes, which are perfectly straight in themselves and plumb, in position, when expanded into the two tube sheets. They are arranged in parallel rows, from furnace

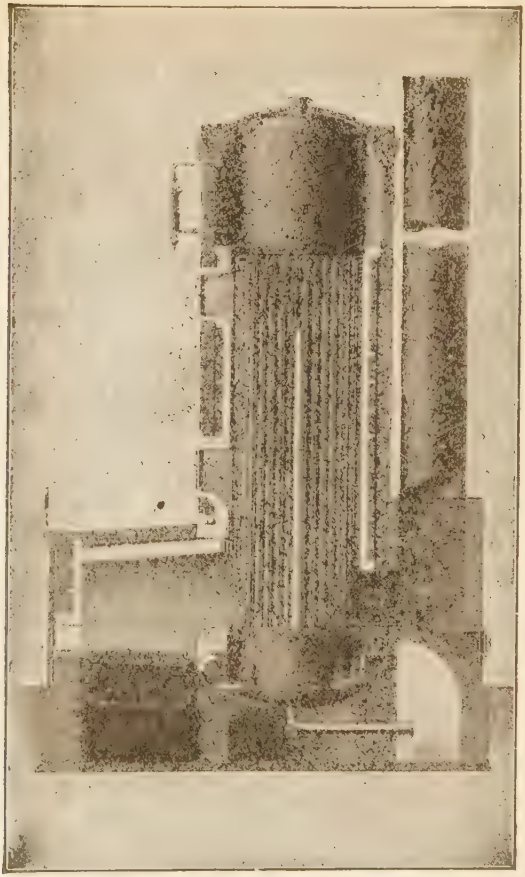


FIG. 18
WICKES VERTICAL WATER TUBE BOILER

to stack, with a clear space between rows sufficient to permit of introducing a small hoe for the purpose of removing

any deposit of soot or of sediment which has fallen from the tubes and accumulated on the mud drum tube sheet.

Manholes. In the convexed head of the steam drum one large manhole, and a number of handholes, are placed, and in the shell of the mud drum on a level with the floor another manhole is placed.

This arrangement permits entering the boiler at the highest and lowest points by simply breaking two joints, from which points an examination may be made, or the tubes cleaned. By the introduction of heavy fire clay tile, the tubes are divided into two compartments. The tubes in the forward compartment are called the "risers," and those in the rear compartment the "downcomers," since the heat, and the water, mingled with steam, rise in the forward tubes, and both heat and the water in a solid column descend in those forming the rear compartment, the steam having passed into the upper drum. This gives the heat two complete sweeps through the entire length of the boiler, and the second sweep from above downward. The heat in its double passage surrounds completely and closely the tubes in both compartments.

Water Line. The water line in this boiler is maintained in the steam drum, at a sufficient height to insure the complete submersion of the tubes.

Baffle Plate. On a level with the water line, and extending over the tubes in the front compartment, is the baffle plate, which deflects the water of circulation rising, commingled with its steam, directly to the "downcomers," and without splashing and spraying the steam room directly above with globules or masses of water.

Liberating Surface. Fully two-thirds of the entire area of steam drum is liberating surface, and, as the liberation

takes place mainly over the "downcomers," it does so in the quietest manner and in the absence of violent ebullition or turmoil.

Steam Room. The large steam room is therefore entirely free from water, and the steam outlet is the topmost point, which is far away from the water line, in large boilers the distance being from six to seven feet. On the other hand, the blow-off is at the very lowest point, and where all impurities are precipitated by gravity, and by separation due to the flow of the water of circulation.

Feed Water. The feed water may be introduced in the steam drum directly into the "downcomers" and far below the water line, or in the mud drum above the precipitated sediment. The latter method, viz., introducing the feed water into the mud drum is to be preferred.

Setting. The brick work setting of the Wickes boiler is so arranged that it is entirely independent of the weight of the boiler, and is therefore free to expand or contract as its co-efficient may dictate, thus allowing the boiler to expand and contract in accordance with the special laws governing its change of form.

The gases of combustion are closely confined to the tubes, after their generation in the furnace, and on their passage to the stack. The direct flow of the heat is, by virtue of the draft over the tile, and down by the shortest possible route, or path of least resistance; while heat of radiation rises naturally and surrounds the steam drum which, as will be seen by reference to Fig. 18, is surrounded by brick work to its top seam. It is claimed by the manufacturers of this boiler that very dry steam is obtained from it, the upper drum acting as a superheater. A damper, either single or double wing, is placed in the setting at the point of

exit of the gases. It is so designed as to allow the quick and easy removal of the ashes when cleaning is going on. The foundation is so designed that by means of a door through the circular brickwork, a man can enter underneath the boiler, examine the blow-off pipe and rivets, and see that the bottom of the mud drum is kept well and heavily painted.

ATLAS WATER-TUBE BOILER.

The design and construction of the Atlas water-tube boiler will be easily comprehended by reference to Fig. 19, which presents a full view of this boiler before setting. It consists mainly of three drums, and two water-legs extending crosswise, and the tubes running longitudinally. The rear water leg is mounted on rollers in setting, in order to allow for expansion, and contraction of the tubes.

An important and original feature in the general design is, that the water-legs are formed by the continuation of the plates of which the shells of the front and rear drums are composed. Thus no flanged plates, or riveted seams whatever are exposed either to the fire or hot gases at any point. The value of this form of construction lies in the fact that it eliminates the possibility of the crystallization of plates and rivets and consequent cracks, and annoying leaks at the seams arising from overheating and unequal expansion and contraction of double metal thickness, due to exposure to flames and furnace gases of high temperature.

Another exclusive and very valuable feature of the design is the arrangement for passing the steam, after leaving the vessels containing water, through a series of superheating tubes, wherein it loses every particle of moisture

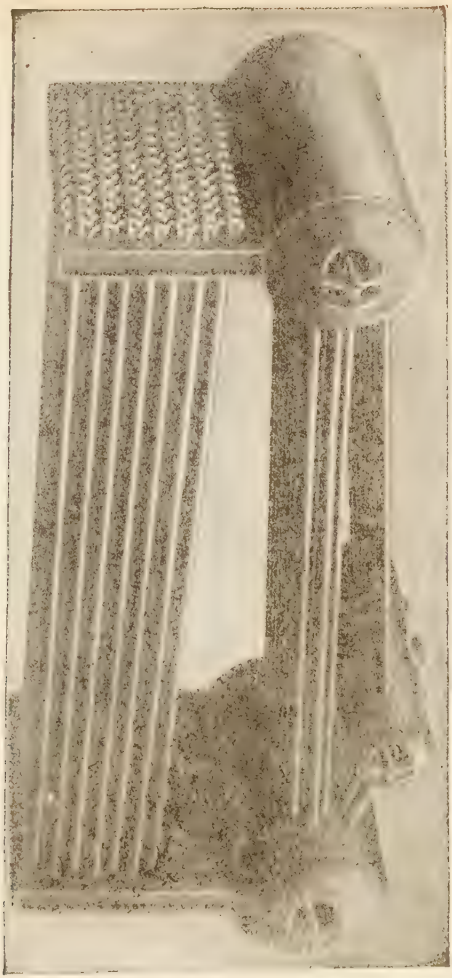


FIG. 19
ATLAS WATER TUBE BOILER

and is heated to a temperature many degrees higher than that normal to its pressure.

Attention is directed to the fact that no matter how large the boiler, there is only one steam drum, consequently the entire product of the boiler can be piped out of a single steam opening. This means a large saving in the cost of piping when compared to the expense incident to installing other types of boilers, which, in units of, say, 200 horsepower, or larger, necessarily have two longitudinal drums to be connected together by the purchaser.

All parts of the Atlas boiler and the fixtures furnished are designed for a safe working pressure of 160 pounds per square inch with a safety factor of more than 5.

All materials have been selected with especial reference to their adaptability to the service required. The shells and heads of the drums and water-legs are open hearth homogeneous flange steel, bearing the maker's stamp of 60,000 pounds tensile strength per square inch of section, with not less than 50 per cent of ductility as indicated by contraction of area at point of fracture under test; an elongation of 25 per cent in a length of 8 inches, and guaranteed by the makers to be capable of bending down flat upon itself when cold, red hot or after being heated to a cherry red and quenched in cold water, without sign of fracture. The chemical and physical properties are determined by thorough tests, and all plates must meet the requirements of the best accepted practice.

The tubes are standard American lap welded, thoroughly tested in all particulars before being expanded into place by roller expanders and again after the boiler is assembled. The tube-holes are accurately placed by template, then reamed and neatly chamfered.

Double-refined iron staybolts and braces, having an ultimate tensile strength of 52,000 pounds per square inch of section, an elastic limit of at least half the tensile strength and an elongation of eighteen per cent in a length of 8 inches, are set to bear uniform tension.

The rivets are of mild steel and can be bent over cold till the sides meet, without developing cracks on the outside of the bent portion. All rivet holes are reamed to perfect fairness, and the riveting wherever practicable is done by hydraulic machinery, upsetting the rivet its full length, completely filling the hole and forming a perfect head in line with it. There are no riveted seams in the fire, or in the path of the furnace gases.

All heads are first heated to a uniform bright red and then flanged at a single operation in a hydraulic flanging press. The bends in the plates forming the bottoms of the water-legs are pressed cold under heavy pressure. There is no distortion, either of heads or plates and there is a total absence of marks resulting from frequent and partial heating and hammering into shape.

No cast metals are used in any parts that are subjected to tensile stresses, or furnace heat. The cast iron that is used for handhole and manhole plates and yokes is good, soft, grey iron, free from flaws or imperfections.

The rapid, steady, unimpeded flow of the water in the course natural to expansion by exposure to heating surface, bears important relation to the most economical utilization of the heat units in the furnace gases, and is an essential factor in the achievement of the highest efficiency in the production of steam. Uniform temperature in all parts exposed to furnace heat, which is so necessary to the safety and durability of the boiler, cannot exist with faulty circulation.

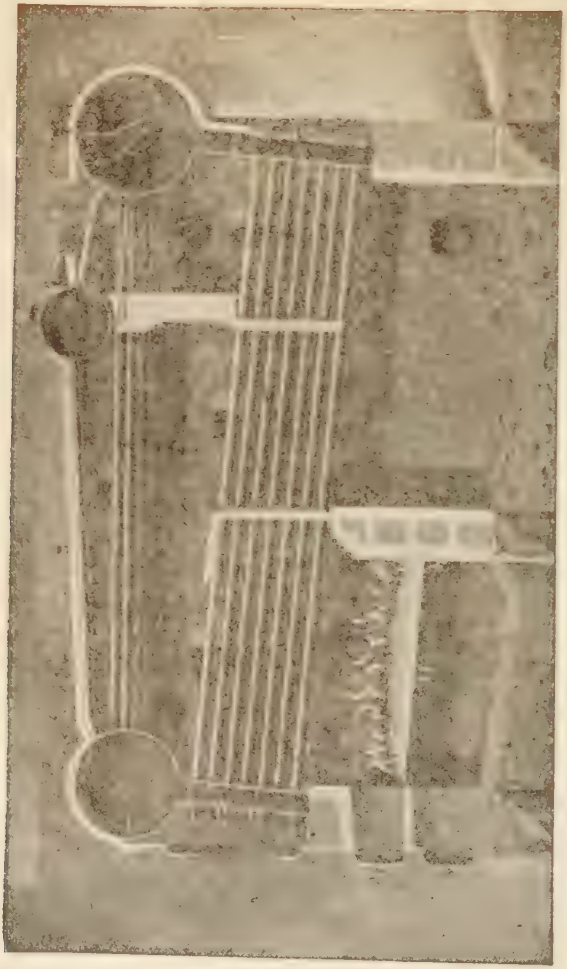


FIG. 20
ATLAS WATER TUBE BOILER—SECTIONAL VIEW

The water is fed into the purifier, whence it overflows into the rear drum and passes down into the rear leg, thence through the inclined tubes to the front leg and upward into the front drum, where the globules of steam generated in the tubes are liberated and carried through the superheating tubes to the steam drum. Meanwhile the water continues its flow through the equalizing tubes to the rear drum, joining the feed current at that point. The regular movement of all the water in the circuit just described, indicates uniform temperature in all the water-tubes, and when an unimpeded circuit has been established in the manner just explained, a maximum supply of dry steam will be delivered. When the circulation is retarded to sluggishness, the water does not so readily absorb the heat, and wet steam and a smaller quantity of it must be expected. It will be noted that while the design of this boiler admits with equal facility the use of either the vertical or the horizontal flame travel, the illustrations herewith show the vertical, it being preferred for several reasons, not the least important of which is that a more uniform distribution of the heat arising from the grates is obtained by first passing the gases upward through the entire nest of tubes between the front end of the furnace, and the first vertical baffle, thence downward between the first and second baffles and finally upward again through the entire nest of tubes between the second baffle, and the rear water-leg. It is not difficult to understand that, aside from all other considerations, the course of the heated gases as described is conducive to a more nearly equal division of the units of heat among all the water-carrying tubes than is possible with the horizontal travel which concentrates by far the greatest degree of heat on the low-

est tubes during almost their entire length. Uniform heat distribution means uniform temperature, which logically followed leads to the reasonable belief that the circulation, which increases or decreases as the temperature goes up or down, is of regular direction in all of the tubes, and that each tube is doing its full share of the work.

Superheated steam has of late years been the subject of much thought and experiment, and now the economical utility of steam containing a number of degrees of superheat is quite generally understood. Ordinary steam at 100 pounds gauge pressure has a normal temperature (omitting decimals) of 338° Fahr., and at 160 pounds its normal temperature is 370° Fahr. The temperature in excess of normal for steam of a given pressure is technically designated superheat. Steam that is in contact with the water from which it was generated cannot be heated above the temperature normal to its pressure. The process of superheating must therefore take place subsequent to the passage of the steam from the vessels containing water. Superheat is obtained by exposing the steam to gases of a higher temperature. The number of degrees of superheat obtainable is governed by the temperature of the gases to which the steam is exposed, and the duration of the exposure. It is claimed for the Atlas water-tube boilers that during various tests under actual working conditions they have produced steam containing from 10 to 30 degrees of superheat.

All users of steam are familiar with radiation losses in steam pipes and the wastes in engine cylinders due to the fact that at each stroke the new supply of steam is brought into contact with the face of the piston and the internal surfaces of the cylinder, which have been cooled

by the exhaust of the previous stroke. When steam is used at normal temperature, each degree of heat thus lost means condensation and a proportional decrease of pressure. When superheated steam is used there is no loss of pressure until the steam is cooled to the temperature normal to the boiler pressure. Up to within the last few years it was customary to equip each boiler with a mud drum. Two very important facts, however, militated so seriously against the mud drum that it is now eliminated from the best boiler practice. In the first place a dangerously large percentage of the substances in the feed water, which at high temperatures become insoluble, would not gravitate toward and settle within the mud drum according to the plan laid out for them, and in the second place, the mud drum proved in many cases little short of an aggravation by reason of the constant and irremediable leakage of the joints between the drum and the boiler, due to greater expansion and contraction of the boiler, the drum being necessarily situated outside of the current of the hottest gases.

Fig. 21, which is a sectional view of the Atlas water-tube boiler as it appears set in brick work, shows a purifying device which is hung loosely by strap hooks inside the shell of the rear water drum, its depth gradually increasing toward the blow-off end.

The water is fed into the shallow end and, the pan being large and always full of, and surrounded by, hot water and steam, it is raised to from 250° to 275° Fahr. before it overflows into the main portion of the drum.

The overflow takes place entirely at the shallow end of the pan, the top of that head being one inch lower than the other head and the sides. It is a well-known fact that water begins to clear itself when it reaches a temperature

of 206° Fahr., and as the liberal dimensions of the pan allow the water to remain in it a considerable time, practically all the scale-forming impurities are precipitated to the bottom of the pan where they remain in a soft sludge-like state pending the opening of the blow-off valve, the frequency of which should be governed by the character of the water and the rapidity of the accumulation.

It will be noted that aside from its open top, which is several inches above the water line in the boiler, the pan is water-tight. It is entirely practicable, therefore, to blow off the sediment as often as desirable while the boiler is under pressure without fear of reducing the water level below the point of safety.

Corrosion is one of the inevitable effects of the accumulation of mud and sediment on metal. Unlike purifiers common to some other boilers, which consist of a pocket-shelf built against the shell or one head of the boiler itself, which, therefore, forms part of the purifier and is exposed to the corrosive action of its contents, the purifier in the Atlas boiler is self-contained and absolutely independent of the boiler-shell or heads. The pan is made in sections and when it finally deteriorates to an unserviceable point, can be removed and replaced through the manhole with little labor and small expense.

It is estimated that an incrustation of $\frac{1}{16}$-inch will cause a loss of 13 per cent of fuel, $\frac{1}{8}$-inch 25 per cent, and so on, and these figures are probably not far from correct. Therefore, the purification of the feed water before it reaches the surfaces exposed to the baking heat of the furnace tends to a more economical use of fuel. It also reduces cleaning labor and the cost of repairs and increases the life of the boiler by avoiding the early disintegration

of the metal which results from subjecting it to that intense heat necessary to boil water through the additional thick-

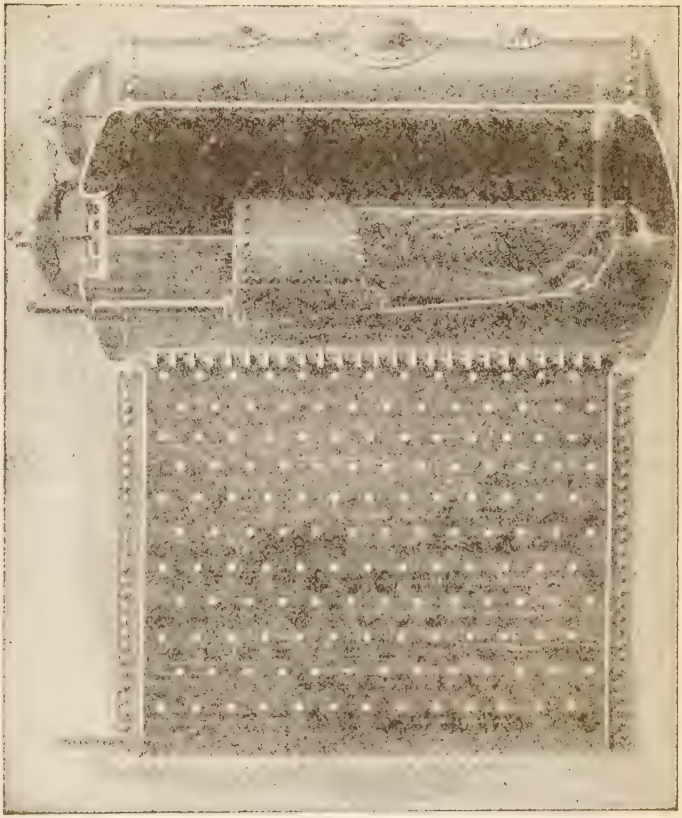


FIG. 21

ATLAS WATER TUBE BOILER

Section through Rear End Showing Water Purifier

ness due to a coating of scale. An individual hand-hole is located opposite each end of each water tube.

These hand-holes are of the diamond-oval shape, having an accurately fitting plate, held in position by a suitable yoke with bolt and nut, the construction being such that the pressure on the inside of the boiler maintains the tightness of the joint. Any one of these plates can be removed and replaced through its own opening without disturbing any of the others. The water tubes in this boiler being absolutely straight, in order to clean them thoroughly on the inside, it is only necessary to remove the front hand-hole plates, insert a scraper and push the sediment back into the rear water-leg, from which it may be easily removed through a few of the hand-holes in the bottom row, or through the blow-off. The internal condition of each water tube may be determined without removing any of the rear hand-hole plates by suspending a light through the full length throat between the rear water-leg and drum, and holding it in turn opposite the rear end of each water tube, while looking through the open hand-holes in front. Or, if the engineer prefers, the top row of rear hand-hole plates may be removed, and the light inserted and suspended, without entering the drum.

The interior of each of the three cross drums of the Atlas boiler is reached through a large manhole in one end. The edge of this manhole forms a deep flange at right angles with the head, is faced true and provided with an accurately fitted plate with yokes, bolts and nuts, all of such proportions that this part of the head is as strong as any other of like area.

MARZOLF WATER-TUBE BOILER.

Another design of water-tube boiler is shown in Fig. 22. The object has been to construct a boiler so that the rela-

tive arrangements of the tubes, drum, and heat passages are such as to obtain the most economical distribution of

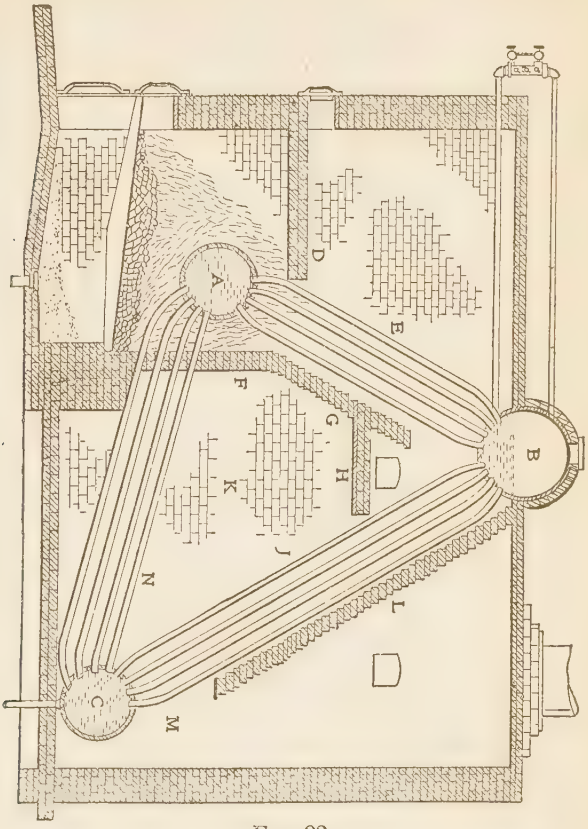


FIG. 22
MARZOLF BOILER—SECTIONAL VIEW

heat. Another object is to facilitate the heating of the water, and increase the circulation by arranging the drums,

and a series of tubes so as to receive the direct application of the heat from the furnace.

As shown, the drum A is located in the furnace and connected to the drum B by tubes slightly bent at each end, the drums A and B being connected to the drum C by similar tubes. As the flame and hot gases rise from the furnace, they surround the water drum A and, striking the arch or baffle wall D, follow along the inclined tubes E, which receive the direct application of the heat generated by the combustion of the fuel. The extension of the bridge wall is reduced in thickness above the grate, as shown at F, forming a back wall which rises vertically behind the water drum A, to a point close beneath the series of tubes E. This wall is surmounted by a sloping wall G, which stands adjacent to, and parallel with, the tubes E, and extends toward the drum B. Owing to the baffle wall H, the hot gases are forced up along the tubes E, and down through the tubes I to the heating chamber K, the roof of which is found by the baffle wall H. The baffle wall L is inclined downward parallel to and close to the rear of the series of tubes J from a point directly behind the steam drum B, to a point over the mud drum C, whereby the heat and products of combustion are deflected downwards among the tubes N.

The hot gases pass from the heating chamber to the passage M between the lower end of the baffle wall and the mud drum C to a draft passage located between the baffle and the rear wall leading to the stack.

The heat thus not only acts directly upon the water-drum A and upon the portions of the tubes N and E, located within the furnace, but also upon the tubes E above the furnace, upon the steam drum B, the tubes J and the mud

drum C. The portion of the tubes N which lies within the heating chamber K also receives heat to some extent, although the heat does not act directly upon them as upon the other tubes.

The feed water supply to this boiler is admitted through the drum C, and circulates through the tubes N, water drum A, tubes E, steam drum B, and tubes J, back to the mud drum, making one continuous circuit. As shown by the illustration, the steam drum B is relatively larger and consequently of much greater capacity than either of the other two drums. It will be noticed that the tubes N are placed on an incline, owing to the mud drum being located on a lower level than the water drum A, a design intended to cause all sediment carried into the boiler to gravitate to the mud drum C, from which it may be readily removed through the usual blow-off.

DUPLEX WATER-TUBE BOILER.

In the Duplex water-tube boiler, Fig. 23, the features which are most strongly emphasized by the designers are: Delivery of steam from the boiler in a superheated condition, without the use of a special superheater; removal of steam from the boiler at a point where there is no ebullition, and elimination of a great many parts of the undulating header type of boiler. The design of the boiler shows absence of stay-bolted surfaces, the drums are not exposed to the direct action of the fire, all seams or rivets are entirely removed from contact with the heat, rigid connections are avoided between parts, and all joints are expanded.

The Duplex boiler consists of two upper steam drums connected by tubes. Short tubes expanded into the bottom

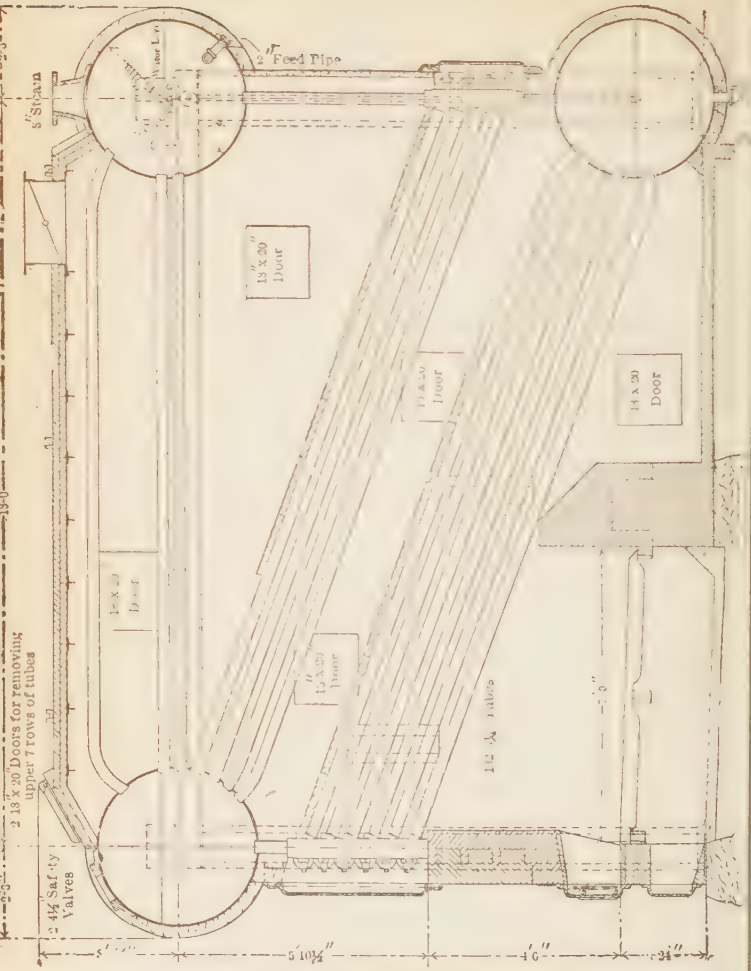


FIG. 23

DUPLEX WATER TUBE BOILER

of the shell of the rear drum form the connection to a set of headers below it, which are connected to the front drum by tubes expanded into the shell of the latter. This comprises the upper generating system. These headers are likewise connected to a drum situated below them by short nipples, and this lower drum is in turn united to a set of headers below, and similarly connected to, the front drum. This comprises the lower generating system. The tubes are inclined 20 degrees to insure rapid and positive circulation.

The drums are made in one sheet with no circular seams except those connecting the heads to the shell. The longitudinal seams are butt-strapped, either double or triple riveted, as the pressure demands, and located on the outside of the shell. A pressed-steel manhole is placed on the circumference of the shell for access to the interior of the drums.

The headers are heavy, made of open-hearth steel, and the section of the header to which the hand-hole plates are secured is designed to be extra heavy, as is that portion where the tubes enter the header, this latter being intended to provide a wide tube seat and obviate the danger of leaky tube joints. The headers are of long box-like form and each header is designed to take in two vertical rows of tubes. An elliptical hand-hole is placed opposite the ends of two tubes through which the tubes are cleaned or removed. This hand-hole is closed by an elliptical cap inside the header held in position by a bolt secured with an outside crab. The joint between the cap seat and the header is made tight with an asbestos gasket to avoid the necessity of remilling the contact surfaces every time a cap is re-

moved and replaced. In this construction the bolt and crab are relieved of the strain of the boiler pressure.

The boiler tubes are 3¼ inches in diameter, of either charcoal-iron, or steel lap-welded, expanded directly into the shell of the drums or into the headers, and the ends are belled over one-quarter of an inch.

A heavy steel framework inclosed in the brick setting supports the boiler. On the sides and independent of the boiler are built walls of brick about 17 inches in thickness. The rear of the boiler is fitted with a sheet-iron casing protected with asbestos covering, the boiler being roofed over with fire tile supported by heavy T-irons. The boiler fronts are of steel and doors of ornamental design are provided for access to the headers. The frame-work is so designed as to provide for the free expansion of all the tubes. The two upper drums are set on lugs secured to the heads, the lugs in turn being supported by the steel framework. The rear lugs are first set on rollers which allow for the expansion of the tubes connecting the two drums. The horizontal style of baffling is used. These baffles rest on the tubes, and guide the gases along the lower bank, and the horizontal circulating tubes.

Finally, the gases pass through the smoke outlet, which may be located on the top of the boiler at the rear, or at a point just under the rear drum. The course of the gases is always upward, which is the free and natural passage for them. The baffles are of the common rectangular shape, which any dealer carries in stock. The upper and lower banks of tubes comprise the active heating surface of this boiler. The tubes that form the lower system are connected at their rear ends to a large mud drum, which acts

as a reservoir for water to insure an ample supply for the bottom tubes at all times. The upper bank of tubes is arranged in reverse of the lower, the tubes being connected to the front drums at their front ends. All the tubes of the upper system discharge independently into this drum. The header ends of both banks of tubes are connected into headers that are straight and of ample area. The tubes are of easy access for the purpose of scraping off soot that has been baked on in service. The feed water is introduced in the upper rear drum, where any air that it contains may be liberated. It then passes downward through the rear circulating tubes and headers to the lower rear drum, where any impurities present in the feed water may be deposited.

The passage of the water is then upward through the lower bank of tubes, through the front headers to the front drum.

At this point any steam that is generated separates from the water, and passes across the steam tubes to the rear drum, and is believed to be thoroughly dried out, and slightly superheated in the process by coming in contact with the hot gases surrounding these tubes. The water passes across through the circulating, horizontal tubes to the rear drum, thence downward again to the rear headers, and thence up the upper bank of tubes into the front drum again. By this time it is expected that the water will have become heated to a very high temperature and, becoming steam, it passes to the rear drum through the superheating tubes, becoming superheated on the route.

From this rear drum the steam is withdrawn, there being no ebullition at this place, as experiment with a boiler with glass heads on the rear drum has shown. The safety

valve is located on the front drum, where its sudden operation cannot throw water into the steam opening. The spaces between the banks of tubes provide access to all parts of the boiler inside the setting through doors in the setting, and other openings in the setting permit the boiler tubes to be cleaned by blowing with steam, or compressed air.

The builders of this boiler, the Robb-Mumford Boiler Company, of South Framingham, Mass., have been carrying on some very satisfactory experimental tests during the past year.

ERIE CITY WATER-TUBE BOILER.

The Erie City Iron Works, of Erie, Penn., has added to its line of products the boiler shown in the accompanying illustrations, Figs. 24 to 26, the reproduced photographs shown being from the experimental boiler at the Erie shops. It is not claimed that the type is novel, but that the Erie iron works will bring to its manufacture and exploitation refinements and improvement in detail and experience and facilities which should soon make a place for it among the standard types.

Unite the three banks of tubes of a Stirling boiler in a single upper drum, placed with its center directly over the center of the lower one, and you have the type. The furnace is an extension on the Dutch-oven plan, allowing great flexibility in the adjustment of grate to heating surface, and introducing the improved furnace conditions of the reverberatory arch. Additional capacity is gained by increasing the length of the drums and the number of tubes sidewise, carrying with it increased width of fur-

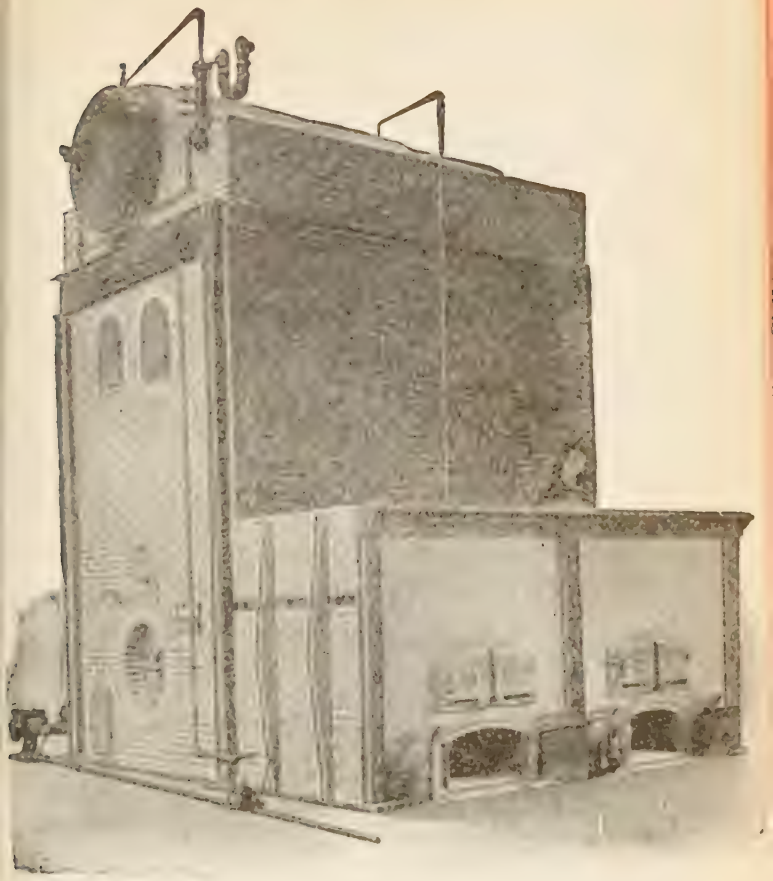


FIG. 24
ERIE CITY WATER TUBE BOILER

nace and proportionate increase of grate surface, while the length of the grate may be made such as to give the desired ratio of grate to heating surface.

The tubes are so spaced that any one of them may be cut out, removed and replaced without interfering with

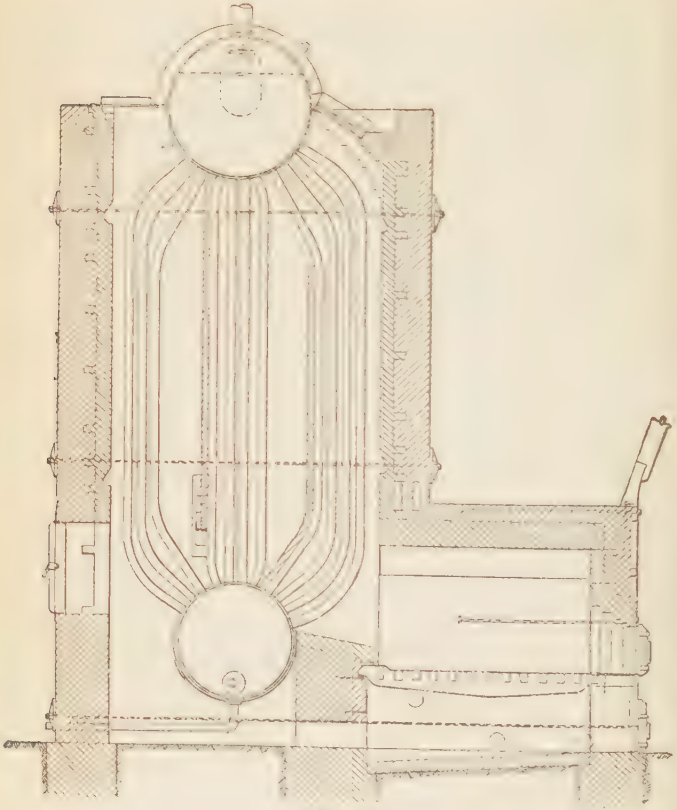


FIG. 25

SIDE ELEVATION ERIE CITY WATER TUBE BOILER

any other. The entire boiler is suspended, as the engravings show, from the upper drum, giving perfect flexibility

and freedom to adjust itself to varying conditions of temperature and stress. The sufficiency of the expanded tube

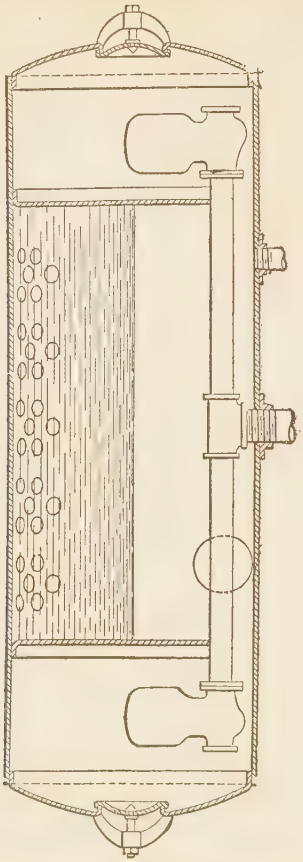


FIG. 26

ENLARGED VIEW SHOWING SEPARATOR IN DRUM OF ERIE CITY BOILER

joints in the upper drum to sustain the weight thus brought upon them has not only been tested out thoroughly in

former boilers of this construction, but has been tried in the boiler illustrated by means of hydraulic jacks and found to be entirely adequate.

In this particular boiler the upper drum is 48 and the lower 40 inches in diameter, with 11 rows of connecting 3-inch tubes and, with 22 tubes in each row, furnishing 2,377 square feet of water-heating surface. The front and rear groups contain four rows each, the central group, three rows.

The baffling is arranged to give three passes as shown, the gases passing longitudinally through each group of tubes. This gives a travel of the gas of something like 10 feet in contact with the heating surface, yet with such freedom of passage that there was little drop in draft pressure between the stack and the furnace when the boiler, nominally rated at 238 horsepower, was developing over 500, and burning 36.7 pounds of coal per square foot of grate.

At each end of the upper drum is a dry chamber, as shown in the longitudinal section (Fig. 26), in which is placed a separator upon each end of the steam-outlet pipe, with the inlet facing toward the end of the drum and away from the steam-liberating surface. The boiler appears to be one which will be well adapted to the large units and intensive service demanded by the modern power plant, especially those in which large amounts of power are required for peak periods and where the ability to stand forcing is particularly desirable.

SETTING RETURN TUBULAR BOILERS.

In setting a return tubular boiler the prevailing custom has been to support on cast iron brackets resting upon the side walls, which are liable in course of time to crumble away and cause trouble. A great improvement is made when we suspend such boilers from I-beams supported by cast iron columns. Figures 27 and 28 show the setting of this type of boilers either singly or in double batteries, by means of suspension. In setting an even number of boilers, as six or eight in one setting, it is best to divide them into pairs so that not more than two boilers will be suspended between supports.

The principal reason for this is that when the large sizes, such as from 150 to 250 horsepower, are used, the size I-beam required to safely carry this load between supports is so large that it overbalances the cost of two or more cast-iron columns.

In setting an odd number of boilers, such as three or five, in a battery, columns are usually placed between the boilers with a 2-inch air space all around the column, and an air duct at the bottom of the setting which runs through from the front to the back and connects with each air space around the column. This allows a free circulation of air, thus tending to keep the columns comparatively cool. In setting boilers in this way, the columns and I-beams are set in position first. The boiler is then hoisted to the proper height by means of tackle, which is attached to the I-beams, and when the boiler is brought to the proper height, the U-bolts are slipped into place and fastened by nuts and washers to the I-beams.

This method abolishes the use of blocking and leaves all of the space under the boilers clear for the brick work.

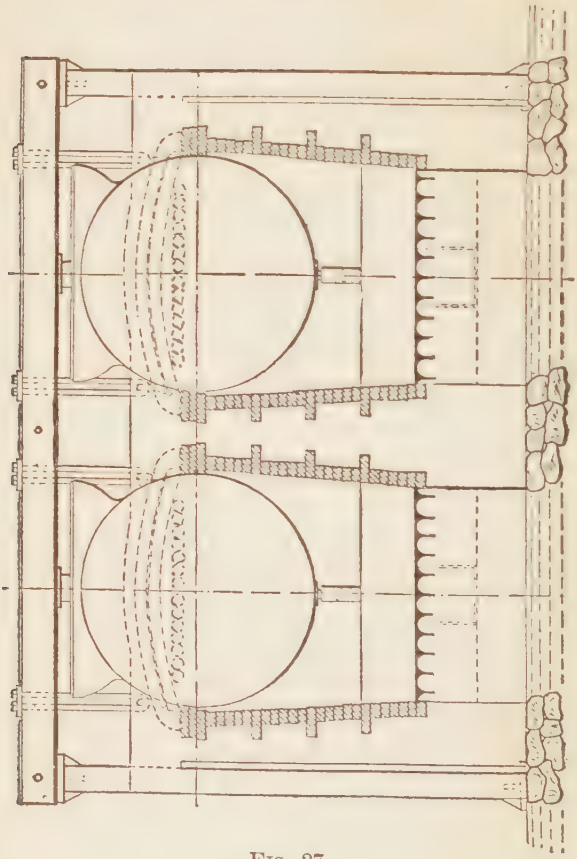


FIG. 27

The expansion is easily taken care of by the U-bolts and hangers, and if the walls are properly set, they will show

no cracks as they carry no weight, and are free to go and come. The accompanying table, No. 1, has been carefully worked out with a factor of safety of 5, and gives the different sizes and lengths of I-beams and columns required

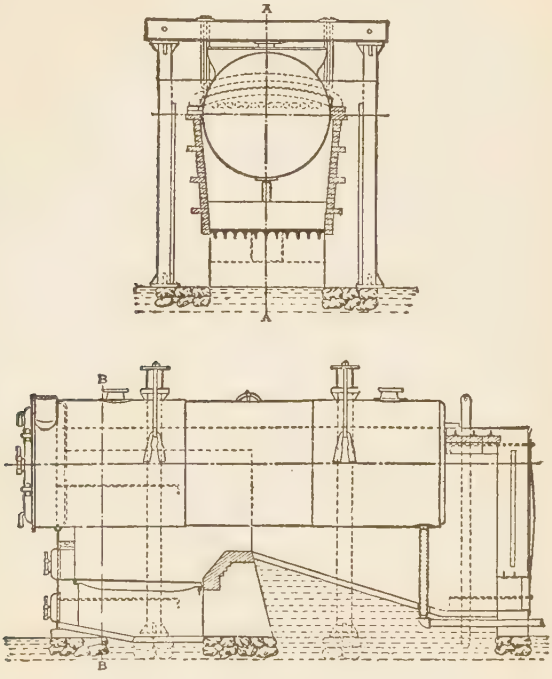


FIG. 28

for boilers of 36 inches in diameter and 8 feet long, to boilers of 90 inches in diameter and 20 feet in length, giving the total weight to be supported and the sizes, weights, and positions of the columns, and I-beams required:

Table 1

SIZES AND WEIGHTS OF COLUMNS AND I BEAMS RE

HORSE POWER	15	20	25	30	35	40	45	50	60
Dia. of boiler in inches	36	36	42	42	44	48	50	54	54
Length of tubes in feet	8	10	10	12	12	12	13	13	15
Length of curtain sheet in inches	11	11	12	12	12	14	14	14	14
Total weight of boiler and water	6500	7500	9400	11500	13500	14200	15500	17800	
Rear head to center of hanger	2-0	2-6	2-6	3-0	3-0	3-0	3-3	3-3	3-9
Center to center of hangers	4-0	5-0	5-0	6-0	6-0	6-0	6-6	6-6	7-6
Front head to center of hanger	2-0	2-6	2-6	3-0	3-0	3-0	3-3	3-3	3-9
Distance between C of supports of boiler	6-0	6-6	7-0	7-0	7-2	7-6	7-8	8-0	8-0
Distance between C of supports of beams	11-8	11-8	12-8	12-8	13-0	13-8	14-0	14-0	14-8
Length of I-beam for 1 boiler	7-0	7-6	7-10	7-10	8-0	8-4	8-6	8-10	8-10
Length of I-beam for 2 boilers	12-6	12-6	13-8	13-8	14-0	14-8	15-0	15-10	15-10
Size of I-beam required for 1 boiler	4	4	5	5	5	6	6	6	6
Size of I-beam required for 2 boilers	6	6	8	8	9	9	9	10	10
Weight per ft. of I-beam for 1 boiler	7.5	7.5	9.75	9.75	9.75	12.25	12.25	12.25	12.25
Weight per ft. of I-beam for 2 boilers	12.25	12.25	18	18	21	21	21	25	25
Length of cast iron col	8-0	8-0	8-0	8-0	8-8	9-3	9-5	10-0	10-0
Outside dia. of C. I. col. for 1 boiler	4	4	4	4	4	5	5	5	5
Outside dia. of C. I. col for 2 boilers	5	5	5	5	5	6	6	6	6
Size of flange on ends of col. for 1 boiler	9½	9½	10	10	10	10½	10½	10½	10½
Size of flange on ends of col. for 2 boilers	10½	10½	12	12	12½	12½	12½	13½	13½
Thickness of C. I. col. for 1 boiler	½	½	½	½	½	½	½	½	½
Thickness of C. I. col for 2 boilers	¾	¾	¾	¾	¾	¾	¾	¾	¾

TABLE 1—CONTINUED.

QUIRED IN SETTING RETURN TUBULAR BOILERS.

70	75	80	90	100	125	150	175	200	200	225	225	250
60	60	60	66	66	72	72	78	78	84	84	90	90
14	15	16	15	16	16	18	18	20	18	20	18	20
16	16	16	17	17	18	18	18	18	20	20	22	22
20800	24800	27200	30300	35000	40000	44000	48000	56000	55000	67000	65000	75000
5 6	3–9	4 6	3 9	4 6	4 6	4 6	4 6	5 6	4 6	5 6	4 6	5 6
7 0	7 6	8 0	7 6	8 0	8 0	9 0	9 0	10 0	9 0	10 0	9 0	10 0
3 6	3 9	4–0	3–9	4–0	4–0	4–6	4–6	5–0	4–6	5–0	4–6	5–0
9 3	9 0	9 6	9 6	9 6	10 0	10 0	10 6	10 6	11 0	11 0	11 6	11 6
16 2	16 2	16 2	17 2	17 2	18 2	18 2	19 2	19 2	20 2	20 2	21 2	21 2
10 0	10 0	10 0	10 6	10 6	11 0	11 0	11 7	11 7	12 0	12 0	12 6	12 8
17 4	17 4	17 4	18 3	18 4	19–5	19–5	20–5	20–6	21–6	21–6	22–6	22–6
7	7	7	7	7	8	8	9	9	9	9	9	10
12	12	12	12	12	15	15	15	15	15	15	15	15
15	15	15	15	15	18	18	21	21	21	21	21	25
31.5	31.5	31.5	40	40	42	42	60	60	60	80	80	80
10 8	10 8	10 8	11 2	11 2	12 0	12 0	12 6	12 6	13 0	13 0	13 10	13 10
5	5	5	6	6	6	6	6	6	6	6	6	6
6	6	6	6	6	8	8	8	8	8	8	8	8
11½	11½	11½	11½	11½	12	12	12½	12½	12½	13½	13½	13½
14	14	14	14½	14½	15	15	16	16	16	17	17	17
¾	¾	¾	1	¾	1	¾	1	1	1	1	1	1
¾	¾	¾	1	1	¾	¾	¾	¾	1	1	1	

QUESTIONS AND ANSWERS.

1. What types of boilers are most commonly used for stationary work?

Ans. The horizontal tubular boiler and the water-tube boiler.

2. Describe in general terms the horizontal tubular boiler.

Ans. It consists of a shell having tubes of small diameter, extending from head to head.

These tubes are located in the water space.

3. What is their function?

Ans. To supply a passageway to the stack for the hot gases from the furnace.

4. Does the water in the boiler receive heat from these tubes?

Ans. It certainly does.

5. Describe the route taken by the smoke and hot gases in the operation of a tubular boiler.

Ans. From the furnace, located under the front end of the boiler, the gases pass under and along the sides of the shell, back to the rear end, the upper part of which is arched over. The route is here reversed, and the products of combustion return through the flues towards the front end and thence through the breeching into the stack.

6. Is this type of boiler economical in the burning of fuel?

Ans. It can be made so if properly set and handled in operation.

7. Describe in a general way the water-tube boiler.

Ans. It consists of a set, or sets of tubes 3 to 4 inches in diameter, sometimes vertical, and sometimes inclined,

and connected at the top to a steam drum, and at the bottom to a mud drum.

8. What advantages as regards circulation of the water has the water tube boiler?

Ans. It provides for a free circulation.

9. Name another advantage connected with the water tube boiler.

Ans. The margin of safety from dangerous explosions.

10. Why is this?

Ans. Because if one or more tubes give way the pressure is relieved.

11. What precautions should be observed in the design and construction of a boiler?

Ans. The best materials should be used, the boiler should be simple in design, and the workmanship should be perfect.

12. Where should the mud drum be located?

Ans. In a place removed from the action of the fire.

13. What should be the capacity of the boiler relative to its work?

Ans. It should have a steam and water capacity sufficient to prevent any fluctuation in either the steam pressure, or the water level, if properly fed.

14. Why should the water in a boiler circulate freely and constantly?

Ans. In order to maintain all parts at as near the same temperature as possible.

15. What should the strength of a boiler be, relative to the strain it is liable to be subject to?

Ans. It should have a great excess of strength.

16. Is a combustion chamber an advantage to a boiler?

Ans. It is, in order to complete the combustion of the gases before they escape to the chimney.

17. How should a boiler be arranged with regard to cleaning?

Ans. All parts should be easily accessible for cleaning and repairs.

18. What type of boiler is the Cahall?

Ans. It is a water-tube boiler.

19. Is it vertical or horizontal?

Ans. It is built either way.

20. What form of Cahall is generally used in central power stations?

Ans. The horizontal form.

20a. What is the range of pressures that these boilers are built for?

Ans. From 160 to 500 pounds per square inch.

21. Describe the method of constructing the joints.

Ans. The sheets are beveled on the edges, bent into shape, and rivet holes drilled after bending.

22. What is gained by so doing?

Ans. Absolutely round rivet holes and no crystallization.

23. What type of riveted joint is used on the higher pressure boilers?

Ans. Triple riveted, double strapped.

24. How are the tubes connected to the steam drum in the Cahall boiler?

Ans. By nipples connected to saddles on the drum.

25. Does this boiler rest upon the brick work?

Ans. It does not, but is suspended free from the masonry.

26. What advantage is there in this style of setting?

Ans. The entire structure is free to expand or contract without causing any strains on either boiler or brick work.

27. Describe the Heine boiler.

Ans. It consists of one, and sometimes two shells or drums resting upon water legs riveted to each end. These water legs are connected by horizontal tubes. The water fills the tubes, water legs, and partially fills the shell, leaving the upper portion for steam space.

28. In the setting does this boiler occupy a horizontal position?

Ans. No. The shell and tubes have an incline of one inch in twelve from front to rear.

29. What provision is made for cleaning and repairing the tubes?

Ans. Hand-holes are located in the head plates opposite each tube.

30. How are these hand-holes closed?

Ans. In the ordinary way, by plates.

31. Where is the mud drum located in the Heine boiler?

Ans. Inside the shell, near the bottom.

32. How is the Heine boiler supported in the setting?

Ans. The front or fixed end rests upon cast iron columns. The rear water leg upon rollers.

33. Describe in brief the Babcock & Wilcox boiler.

Ans. It is composed of wrought iron tubes, placed in an inclined position, and connected with each other, and with a horizontal steam, and water drum by vertical headers.

34. Where is the mud drum in this boiler?

Ans. In the rear, and connected to the lowest part of the boiler.

35. What provision is made for cleaning the tubes in the Babcock & Wilcox boiler?

Ans. Through hand-holes in the headers, opposite each tube?

36. How is this boiler supported in the setting?

Ans. It is suspended from wrought iron girders, entirely independent of the brick work.

37. Describe in general terms the Stirling boiler.

Ans. It consists of three upper steam drums, each being connected by a number of tubes to a lower or mud drum.

38. How are the steam spaces connected?

Ans. By shorter tubes.

39. How is the boiler supported?

Ans. On a structural steel frame work.

40. What provision is made for expansion and contraction of the tubes?

Ans. They are slightly curved near the ends.

41. How are the hot gases directed in their course from furnace to stack?

Ans. By means of fire brick baffle walls.

42. How is the interior of this boiler cleaned?

Ans. Four manholes are provided in the drums, by which access to the interior of both the drums and tubes is obtained.

43. What type of boiler is the Maxim boiler?

Ans. It is a water-tube boiler consisting of two drums, one above the other, connected by tubes.

44. Describe the tubes.

Ans. Each tube has two bends, thus providing for unequal expansion or contraction.

45. How is the heating surface of the Maxim boiler arranged?

Ans. It is so arranged that the current of heated gases is made to travel three times the length of the tubes, the direction of the current being changed seven times in its route from furnace to stack.

46. What can be said of the Bigelow-Hornsby water-tube boiler?

Ans. Owing to the flexible form of its construction it is possible to build it in very large units, 2,000 horsepower and upwards.

47. What peculiar feature makes this possible?

Ans. Each section is independent of its neighbor, except the nipples connecting with the steam drum, and the equalizing nipples connecting the bottom drums of the rear sections.

48. How is the boiler supported?

Ans. Entirely from overhead beams.

49. What percentage of the heating surface do the tubes of the front unit comprise?

Ans. More than 12 per cent.

50. Where is the feed water first admitted?

Ans. Into the bottom drum of the rear unit.

51. Describe the course of the feed water.

Ans. The feed water is admitted into the bottom drum of the rear unit, and is advanced gradually from the coldest to the hottest portion of the boiler.

52. How is the speed of the feed water up the rear unit regulated?

Ans. By the amount of steam generated, ample time being permitted for scale forming matter to be deposited in the bottom drum of this unit.

53. Where does the liberation of steam take place?

Ans. In the upper drum.

54. What can be said of this boiler regarding the utilization of the heat?

Ans. It is baffled so that the products of combustion are carried uniformly over the heating surfaces in thin layers, the baffle plates serving to guide the gases through in substantially uniform passages.

55. To what factor of safety is the Bigelow-Hornsby boiler built?

Ans. Five for 200 pounds working pressure.

56. Describe in brief the Wickes vertical water-tube boiler.

Ans. It consists of two cylinders joined together endways by straight tubes, and erected in a vertical position.

57. What can be said of the top cylinder?

Ans. It is the longer, and is designated the steam drum.

58. What about the bottom cylinder?

Ans. It is the shorter, and is designated the mud drum. Both cylinders vary in dimensions as to diameter and length, according to the power required of the boiler.

59. Where are the manholes of the Wickes boiler?

Ans. One is placed in the convex head of the steam drum; there are also a number of hand-holes in this head. A manhole is also placed in the lower or mud drum, near the floor, thus permitting access to the top and bottom of the boiler.

60. How are these tubes divided?

Ans. By heavy fire-clay tile these tubes are divided into two compartments. Those tubes in the front compartment are called the "risers" and those in the rear the "downcomers."

61. What can be said of the heat in its double passage?

Ans. It surrounds completely, and closely the tubes in both compartments.

62. Where is the water line in this boiler?

Ans. At a sufficient height in the steam drum to insure the complete submersion of all the tubes.

63. How is the brick work setting of the Wickes water-tube boiler arranged?

Ans. It is independent of the weight of the boiler, and free to expand or contract.

64. Describe briefly the design of the Atlas water-tube boiler.

Ans. It consists mainly of three drums and two water legs extending crosswise, while the tubes extend lengthwise.

65. What is the original feature in the design of the water legs?

Ans. They are formed by the continuation of front and rear shell plates.

66. What other valuable feature is claimed for this boiler?

Ans. After the steam leaves the vessels containing water it is passed through a series of superheating tubes, and is superheated.

67. Describe the course of the feed water.

Ans. It is fed first into the purifier, whence it overflows into the rear drum and down into the rear leg, thence through the inclined tubes to the front leg, thence up into the front drum, where the steam is liberated and carried through superheating tubes to the steam drum.

68. What are the facilities for cleaning the water tubes of this boiler?

Ans. An individual hand-hole is located opposite each end of each water tube.

69. How is the interior of each of the three cross drums reached?

Ans. Through a large manhole in each end.

70. Describe briefly the design and construction of the Marzolf water-tube boiler.

Ans. It consists of three drums connected with each other in triangular form. Drum A directly over the fire is connected by tubes with drum B above it, and with drum C in the rear and slightly below it. Drum C, which is the mud drum, is also connected with drum B. The tubes are each slightly bent. The steam is collected in drum B, which is maintained about one-third full of water.

71. Describe in brief the action of the heat upon this boiler.

Ans. It acts first upon the water in drum A over the furnace, then by means of a baffle wall it is carried along the inclined tubes to drum B, where it is deflected and carried down along other inclined tubes to drum C, thence to the stack.

71. How are the products of combustion caused to act upon the lower bank of tubes?

Ans. By means of baffle walls located in the rear of the furnace.

72. At what point in this boiler is the feed water admitted?

Ans. At the lowest point, viz., the mud drum.

73. What are the principal advantages claimed for the Duplex water-tube boiler?

Ans. Delivery of superheated steam; the removal of steam from the boiler at a point where there is no ebullition; the drums not exposed to the direct action of the fire.

74. Describe in brief the design of this boiler.

Ans. Two upper steam drums connected by tubes, a mud drum at the bottom and rear which is connected to the upper drums by headers and short nipples. The tubes are inclined 20 degrees to insure rapid and positive circulation.

75. How is this boiler supported?

Ans. Upon a heavy steel framework.

76. What is the leading feature in connection with the Erie City water-tube boiler?

Ans. The three banks of tubes are practically vertical, connected to upper, and lower drums, and spaced so that any one of them may be cut out for repairs without interfering with the others.

77. How do the products of combustion act upon this boiler?

Ans. The baffling is arranged to pass three times across the tubes, and at each end of the upper drum is a dry chamber.

78. Describe in brief the best method of supporting horizontal tubular boilers.

Ans. By means of hangers suspended from I beams, supported by cast iron columns. This takes the weight off the side walls.

SUPERHEATED STEAM

The properties of superheated steam approximate those of a perfect gas, and its thermal conductivity is lower than that of saturated steam. Higher efficiency and much better working conditions are attainable by the use of superheated steam, and this is especially the case where a large volume of steam is used for the development of power in reciprocating engines and steam turbines. Two main reasons may be given for the use of superheated steam, and these are:

(1) There is always a loss of heat by radiation from steam pipes, and the heat so lost represents an equivalent condensation when the pipe conveys saturated steam. Superheated steam cannot condense; it must first lose all of its superheat and be reduced to saturated steam. In consequence, if sufficiently superheated it can lose the amount of heat represented by radiation from the steam pipes, yet reach the engine perfectly dry. Since the thermal conductivity of superheated steam is less than that of saturated steam, the heat will not be so rapidly transmitted from the body of the steam to the walls of the pipe.

(2) In an engine the steam is admitted into a space which has been cooled by the steam exhausted during the previous stroke. The heat necessary to warm the cylinder walls from the exhaust temperature to the temperature of the entering steam can be supplied only by the entering steam, hence if it be saturated some of it must condense. The amount thus

condensed is seldom less than 20 to 30 per cent of the total weight of steam entering the cylinder. It is obvious, however, that if an amount of heat more than sufficient to warm the cylinder walls could, by means of superheating, be imparted to the steam before it reached the engine, then even after the cylinder walls had been warmed up the steam would remain dry, and the initial condensation would thus be overcome.

These properties of superheated steam have long been known, but their practical application has been slow, owing to constructive difficulties. The recent successful development of the steam turbine, and of reciprocating engines adapted to the use of superheated steam, has rendered necessary the development of a class of simple, durable, efficient and safe steam superheaters. With reciprocating engines the presence of moisture in the steam increases cylinder condensation, whereas by the use of a superheater, dry steam may be obtained, and an increase of 12 to 20 per cent in economy is effected with a superheat of only 40° to 100° F.

Steam turbine practice shows for a superheat of 0° to 100° F. 1 per cent improvement in the steam consumption for each 10° superheat, and for a superheat of 100° to 200° F., 1 per cent improvement in steam consumption for each 12° superheat.

Another important point connected with the use of superheated steam, especially with steam turbines, is that efficiency increases considerably with the increase in superheat, as shown by the following figures: For 100° superheat, 2.5 per cent better efficiency, and for 200° superheat 6.75 per cent better efficiency. The saving in coal due to the use of superheated steam is not as great as is the saving in steam consumption.

For instance, a superheat of 100° F. will reduce

the steam consumption of a turbine 10 per cent. Assuming the temperature of the feed water to be 200° F., the heat absorbed in the boiler for one pound weight of steam at 150 pounds gauge pressure without superheat is 1027 B. T. U.'s. The heat in 0.9 pound weight of steam under the same conditions is 924.3 B. T. U.'s, a saving of 102.7 B. T. U.'s. Now in order to effect this saving, a superheat of 100° F. must be applied to each pound weight of steam used. This will require 57 B. T. U.'s for one pound weight, or 51.3 B. T. U.'s for each 0.9 pound weight of steam used. This amount of heat added as superheat saves twice as much heat in the boiler. As a general proposition then it may be stated that a fuel economy equal to one-half of the saving in steam consumption is effected by superheated steam.

The proper temperature to which the steam in a given plant should be superheated depends upon that degree of superheat for which the engines or turbines are best adapted, and this can only be determined by trials and tests; but having been once determined it should then be maintained continuously. A superheat lower than the mean will cause a loss in economy, while a superheat much higher than the predetermined mean will cause trouble with valves and fittings.

Early superheating specifications sometimes called for a superheat of 300° F., and an excessive amount of superheating surface. Cast iron fittings, however, were ruptured at these high temperatures, and the added cost of cast steel fittings more than counterbalanced the gain in economy due to the use of superheat. The highest commercial efficiency has, as a result, been found to be obtained by the use of a moderate superheat, and modern installations now generally use a

superheat of from 125° to 150° F. in turbine plants, and the same or a little lower superheat in engine plants, where a low superheat eliminates moisture and reduces cylinder condensation without causing lubricating troubles.

The temperature to which the steam will be raised will depend both upon the quantity and temperature of the steam flowing through the superheater and upon the temperature and quantity of the gases flowing over the superheater tubes. As the quantity of steam is determined by the boiler load, the temperature will be fixed by the temperatures and quantity of the gases. These will depend on the design and arrangement of the boiler and superheater.

Construction and Installation of Superheaters. The superheater should be constructed to admit of all its parts expanding and contracting freely, without severe strains being put on any of the joints which might cause them to leak. Flanged joints, when exposed to furnace gases, should be avoided wherever possible.

Great care should be exercised in installing a superheater to see that it is placed in a position where a suitable temperature is available. If a boiler is working under economical conditions, the difference between the temperature of saturated steam and of the escaping flue gases leaving a boiler is not sufficient to superheat the steam produced, unless the surface of the superheater be very large; hence, generally speaking, a superheater should not be placed in the flue between boiler and chimney. On the other hand, if placed too near to the boiler furnace, or in a separate furnace, considerable temperature fluctuations may occur, and possibly an excessive temperature may be attained which would cause damage.

The particular method of installation to be pursued in a given case will depend upon the type of boilers with which the plant is equipped. If of the horizontal return tubular type, the superheater may be located in the back connection, in which case it will be in the immediate path of the hot gases on their way from the furnace to the back ends of flues. The method usually pursued in the installation of a superheater in connection with water tube boilers is to place it in the path of the gases of combustion after these gases have passed over a portion of the boiler or tube surface, but there are objections to this method owing to the fact that the first pass of the gases over the heating surface of a correctly designed water tube boiler absorbs 75 to 80 per cent of the heat in the gases, thus bringing their temperature comparatively low, and unless the superheater surface is quite large, it becomes difficult to obtain adjustment or close regulation. Various methods have been devised for controlling the volume of gas passing over the superheater surface. In some instances this is accomplished by means of dampers; in other cases the superheater is located in a separate flue; and in some instances where the demand for superheated steam is imperative, the superheater is provided with a separate furnace. Still another method of temperature regulation is illustrated in Fig. 29, which shows the Heine superheater with a portion of the header broken away to show the interior construction. It will be noticed that the header box is divided into three compartments, so that the steam makes four passes through the bent tubes on its way to the engine or turbine. When installing a superheater, provision should always be made to protect it against overheating when steam is not passing

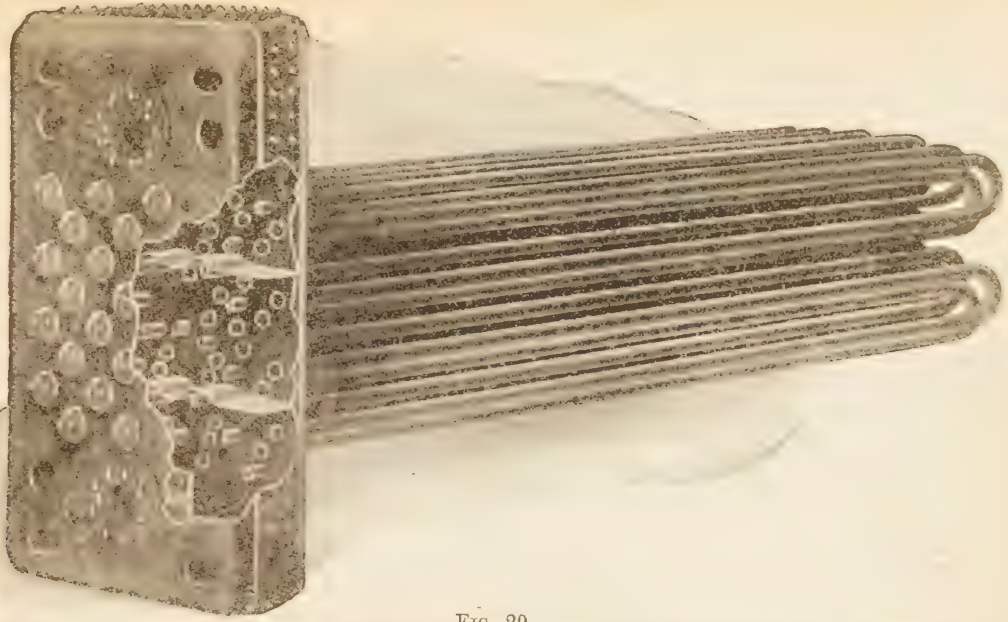


FIG. 29

THE HEINE SUPERHEATER, SHOWING HEADER CONSTRUCTION AND DIAPHRAGMS WHICH DIRECT THE STEAM THROUGH THE TUBES IN FOUR PASSES

through it in sufficient quantity to carry away the heat supplied by the furnace gases.

In some cases this is accomplished by connecting a flooding pipe to the superheater so that it will be

FIG. 30
INLET AND OUTLET STEEL HEADERS OF FOSTER SUPERHEATER, SHOWING METHOD OF ATTACHING ELEMENTS

possible to fill it with water, either from the boiler or other source during the time that steam is not passing and the superheater is subject to the action of the

furnace gases at a high temperature. Such a condition can exist when the load is temporarily arrested, or when getting up steam in a boiler preparatory to putting it into service. However, there are objections

FIG. 31. RETURN BEND ELEMENT USED IN CONSTRUCTION OF THE FOSTER SUPERHEATER SHOWING PROTECTING COVERING AT BEND

to this practice. The admission of water to the interior of the tubes or superheater elements which are at the same time receiving heat from the furnace gases

is likely to result in the formation of scale within the tubes, which in time will cause a decrease in efficiency and also stoppage of circulation.

Another serious objection to flooding is the necessity of having to open and close certain valves on each boiler, which operation requires to be reversed when the critical period is passed, and the superheaters are to resume their normal function. When a superheater has been flooded it must be drained before being put back into service and, unless this operation is performed with the utmost care and intelligence, disastrous results are likely to follow, due to the water in the superheater being driven into the steam main.

From the foregoing it would appear that the ideal in superheater construction requires the absence of any form of connection between the water space of the boiler and the superheater, and that some other means besides flooding be provided for protection against overheating. Various designs of superheaters have been produced with this object in view, among which may be mentioned the Foster superheater, in which the elements consist of seamless drawn steel tubes, on the outside of which are fitted a series of cast iron annular gills or flanges placed close to each other as shown in Figs. 30 and 31, carefully fitted to the tube so as to be practically integral with it. This construction exposes an external surface of cast iron to the action of the hot furnace gases, and the protection thus afforded is ample to prevent damage to the superheater during the process of steam raising, or when the normal passage of steam is temporarily arrested and thus renders it possible to dispense with all flooding devices. Fig. 32 shows the Foster superheater installed in connection with a battery of horizontal

water tube boilers, the superheater being located in the space between the water tubes and the drum, but having no connection whatever with the water space of the boiler. When attached to boilers of the hori-

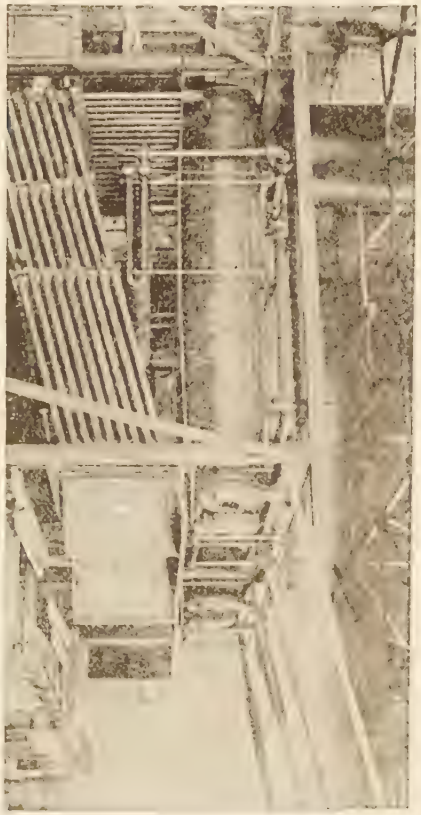


FIG. 82
FOSTER SUPERHEATER COMBINED WITH A FIRE-TUBE BOILER

zontal return tubular type, the Foster superheater is located in the rear connection directly in the path of the furnace gases.

OIL FUEL FOR STEAM BOILERS

The possibility of substituting oil for coal as fuel for steam boilers has been the subject of much study by engineers since the supply of fuel oil has become fairly abundant and dependable. In some sections, as for instance the Pacific Coast, the use of oil as fuel has been largely standardized, but in the eastern states coal is still the standard and likely to remain so for some time.

Advantages and Disadvantages of Oil Fuel. The advantages of oil fuel may be summarized as follows:

(1) Heat value per pound at least 30 per cent higher than that of high-grade coal.

(2) The space required for storage of a given weight of fuel oil is much less than that required for an equal weight of coal, and if heating values be compared, 50 per cent more heat can be stored in oil than would be contained in an equal number of cubic feet of coal.

(3) A great saving in the labor of handling oil fuel. Entire absence of ashes.

(4) Instead of losing heat by storage, as coal does, oil maintains its heating qualities indefinitely provided the storage tanks are equipped with ordinary ventilation.

(5) Absence of the dust and dirt unavoidable with coal. Consequently the boiler room and equipment of pumps and other machinery may be kept clean. A lower temperature can also be maintained in the boiler room.

(6) No firing tools required, such as coal shovels, slice bars, ash hoes, etc.

(7) Ease of regulation whereby a constant steam pressure can be maintained.

(8) The heat is more evenly diffused over the entire heating surface of the boiler, therefore the strain on the sheets and rivets is lessened.

(9) A higher boiler efficiency due to (a) more perfect combustion with less excess air; (b) more equal distribution of heat in the combustion chamber due to the fact that the fire doors remain closed, and (c) small amount of soot deposited.

(10) An increase in boiler capacity of 35 to 50 per cent, depending upon grade of coal and draft conditions.

(11) A great reduction in required stack area for boilers using oil fuel, as compared with an equal boiler capacity operated on coal. In other words, boilers fired with oil require only 60 per cent as much stack area as would the same boilers if operated to an equal capacity with coal as fuel.

(12) Owing to cleaner conditions of tubes and air ducts, and to the smaller volume of air which has to pass through the furnace for a given calorific capacity of fuel, less heat is lost up the stack with oil as fuel than there is with coal.

The disadvantages attendant upon the use of oil fuel for boilers are as follows:

(1) Low flash point, which is the temperature at which inflammable gas or vapor is given off. The flash point of fuel oil should be not lower than 140° F.

(2) Conflict with underwriters' or city requirements which specify that storage tanks for fuel oil be located underground and at least thirty feet from the nearest building. This can generally be complied with in the average power plant, except in the congested

districts of a city, where it is likely to be prohibitive.

(3) The intense temperature developed in the furnace may tend to increase the deposit of scale in boilers using feed water impregnated with scale-making ingredients, although with a proper setting for burning oil, and the exercise of care and good judgment in its manipulation, repairs due to this source should be less than with coal.

Heat Value. Fuel oil depends for its heat value almost entirely upon hydrogen and carbon, which although present in the liquid in many different combinations, still do not vary greatly in relative proportions when reduced to ultimate analysis. Thus, the hydrogen content of various crude oils varies usually from about 11 or 12 per cent to about 14 per cent, and the carbon from about 84 to 87 per cent: slight amounts of oxygen, nitrogen and sulphur also are present.

The ultimate analysis may serve for calculating the heat value of a fuel, but the use of the bomb calorimeter of the Mahler type is universally accepted as standard. In this instrument the latent heat of the vapor formed by the combustion of the hydrogen is included in the result—this proportion of the heat value is, however, not available for use in a furnace where the vapor passes away as superheated steam.

The heat value of fuel oil varies between about 16,000 and 20,000 heat units per pound, 18,000 to 19,000 being the common values.

The following tables gives the computed values in British Thermal Units (B. T. U.'s) per pound, also the specific gravity, weight per barrel, and B. T. U.'s per barrel of California oils, as determined in the laboratories of California, and furnished through the

courtesy of Prof. Edmond O'Neil and his assistants. From this table it will be seen that, although the average heating value per pound of crude oil increases as the specific gravity diminishes, it does not increase so rapidly as the weight per unit of volume diminishes. The heating value per barrel of the heavier oils is therefore greater than that of the light ones.

SPECIFIC GRAVITY, WEIGHT AND HEAT VALUE OF OIL

(Union Oil Company)

Deg. Baume.	Specific Gravity.	Weight per bbl.	B. T. U. per lb.	B. T. U. per bbl.	Deg. Baume.
10	1.0000	350.035	18,280	6,398,600	10
11	0.9929	347.55	18,340	6,374,100	11
12	0.9859	345.10	18,400	6,349,800	12
13	0.9790	342.68	18,460	6,325,900	13
14	0.9722	340.30	18,520	6,302,400	14
15	0.9655	337.96	18,580	6,279,300	15
16	0.9589	335.65	18,640	6,256,500	16
17	0.9524	333.37	18,700	6,234,000	17
18	0.9459	331.10	18,760	6,211,400	18
19	0.9396	328.89	18,820	6,189,700	19
20	0.9333	326.69	18,880	6,167,900	20
21	0.9272	324.55	18,940	6,147,000	21
22	0.9211	322.42	19,000	6,126,000	22
23	0.9150	320.28	19,060	6,104,500	23
24	0.9091	318.22	19,120	6,084,400	24
25	0.9032	316.15	19,180	6,063,800	25

The specific heat of oil varies with its composition. It will be greater the richer the oil is in hydrogen, and lower in proportion to a greater carbon content.

The following figures are reproduced from Holde's work on Examination of Hydrocarbon Oils:

Crude oils from—	Sp. Gr.	Sp. Ht.
Japan	0.862	0.453
Pennsylvania	0.810	0.500
Russia	0.908	0.435
California	0.960	0.398
Bustenari	0.842	0.462

Viscosity. As the problem of handling fuel oil and burning it to advantage in boiler furnaces hinges to a large extent upon the viscosity of the oil, it may be well at this point to explain the meaning of the term. Viscosity is the resistance to internal movement, or the internal friction of a liquid. It may be measured in various ways, but the most common method is by noting the length of time required for a definite quantity of the liquid to pass through an orifice, or short pipe under known conditions of temperature and head. It is on this principle that the best known types of viscometers are constructed.

In atomizing oil by means of the mechanical atomizer or pressure burner it is important to know to what degree the viscosity must be reduced and to what temperature the oil must be heated.

The Flash Point, already referred to, is that degree of temperature at which the oil will give off an inflammable gas or vapor, and it is determined by simply heating the oil, and as the temperature rises, testing it with a spark or flame until the distilled vapor ignites and a flash is apparent. Certain oils require heating above the flash point in order to reduce the viscosity to a point where the mechanical atomizer will properly perform its function.

Atomization and Burning of Fuel Oil. Thermometers should be used on both suction and supply pipes so that the fireman will at all times know the temperature of the fuel. It is necessary to have an unvarying pressure of steam or air through the burner to atomize the fuel. Also, the air used to support combustion should be delivered through a nozzle under the burner at a constant pressure. The oil pressure must not fluctuate; under no circumstances should it vary more than

one pound. The oil pump, which should be brass lined, with the aid of the pulsometer and superheater, will insure a constant circulation of fuel to the various burners, maintaining an unvarying pressure on the supply pipes. All oil pipes should be of adequate size and carefully laid.

A non-carbonizing burner of modern construction, that will externally atomize the fuel and distribute the heat over the entire width and length of the furnace or firebox, should be used. If steam is used through the burner to atomize the oil, it should be taken from the top or dome of the boiler or superheater so that it will be as dry as possible. The atomizer orifice in the burner should be as small as practicable to provide for the minimum and maximum capacity of the burner.

The requirements for the perfect combustion of liquid fuel are as follows: Reduction to a fine spray or complete atomization; bringing it into contact with the proper amount of air; mixture of oil spray and air burned in a furnace of a refractory material with room enough to complete combustion before the gases come in contact with the boiler heating surfaces. The first condition is fulfilled by selecting a proper burner, and the remaining conditions can generally be obtained by making slight changes in, or additions to, the existing furnaces.

The question whether to use steam or air for atomizing the oil seems generally to have been decided in favor of steam, as experimental results show that it takes about the same amount of steam to operate the air compressor as it does to atomize the oil at the burner and the additional investment and complication involved with greater possibility of interrupted service is avoided.

Types of Oil Burners. Although thousands of patents have been granted for oil burners or atomizers, two general subdivisions, or five general classes, as designated by the U. S. Naval Liquid Fuel Board, cover practically all of the main features of construction. The two general subdivisions are:

(a) Outside mixing. Oil and atomizing agent meet outside the burner.

(b) Inside mixing. Oil and atomizing agent meet inside the burner.

The five general classes are:

(c) Drooling. Oil oozes out on to steam or air jet. (See Fig. 33.)

(d) Atomizing. Oil is swept from orifice by steam or air jet.

(e) Chamber. Oil mingles with steam or air in body of burner and the mixture issuing from nozzle is broken into minute particles by the expansion of the steam. (See Fig. 34.)

(f) Injector. Similar in principle to boiler feeding injector.

(g) Mechanical spraying. Effected by mechanical means without the use of atomizing agents such as steam or compressed air.

The important features which should be embodied in all burners are, easy method of installation, construction that will allow quick inspection, easy removal of all foreign materials which may clog the burner at any point, and rapid and cheap renewal of any parts which are subject to wear.

Fig. 33 shows a side sectional view of one form of outside burner in which the oil, as it oozes slowly out of the top passage is immediately brought into contact with a flat, thin jet of steam issuing from the lower

opening which is in the form of a long, narrow slot. The result is that the oil is thoroughly atomized and the hot flame completely fills the furnace. A view of

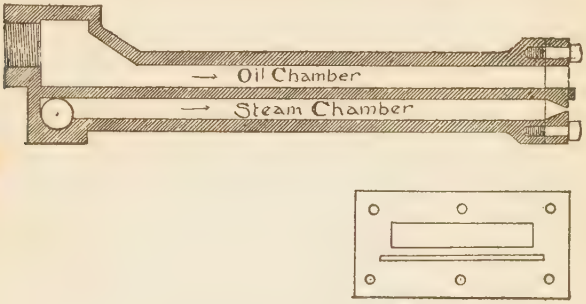


FIG. 33
OIL BURNER—OUTSIDE MIXING

the front plate showing the oil and steam openings as they appear in the furnace is given in the lower right hand corner of Fig. 33. A good burner of this type

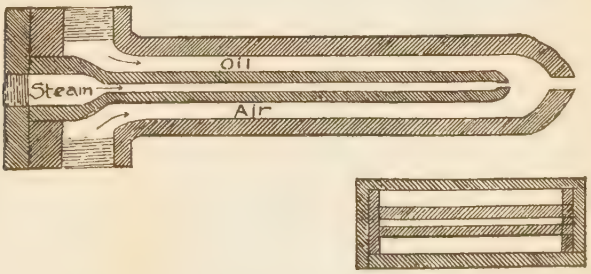


FIG. 34
OIL BURNER—INSIDE MIXING

properly operated should not use over 0.35 pounds of steam per pound of oil atomized.

Fig. 34 is a side sectional elevation of another type of burner or atomizer. This burner properly belongs

in class e, subdivision b, already referred to. It will be noted that there are three inlets to and three passages through, this burner. One passage is for oil, one for steam, and one for air, and all three open into a common chamber called the mixing chamber. From thence the atomized mixture is expelled through the nozzle, into the furnace where complete combustion takes place.

Mechanical Atomizers. In this type of burner the use of steam or compressed air as atomizing agents is dispensed with, and the oil is given a rapid whirling motion in a chamber close to the outlet of the device from whence it is forced under pressure through a small orifice concentric with the axis of rotation. The centrifugal force thus developed sprays the oil in the form of a hollow cone of atoms into the furnace. In order to effect a successful and complete atomization of the oil by means of the mechanical atomizer or spray burner as it is sometimes called, two very important factors must not be lost sight of. These factors are (a) The oil must be forced into and through the burner under a heavy pressure by the oil pump; (b) In addition to being heated to the temperature required to insure a free flow of oil through the pump suction, it is also necessary that the oil after passing through the pump, and before entering the atomizer, be reheated, this time to a temperature of about 270° F. in order that there may be no hindrance to the operation of the burner due to viscosity of the oil, since with viscous oil the plant would be practically inoperative.

Fig. 35 shows an interior view of the Koerting mechanical atomizer, in which the oil tube is surrounded by an annular passage which contains a screw blade

or spiral as shown. This spiral imparts a gyratory motion to the air which is being forced through it under a slight pressure, and this motion is maintained by the

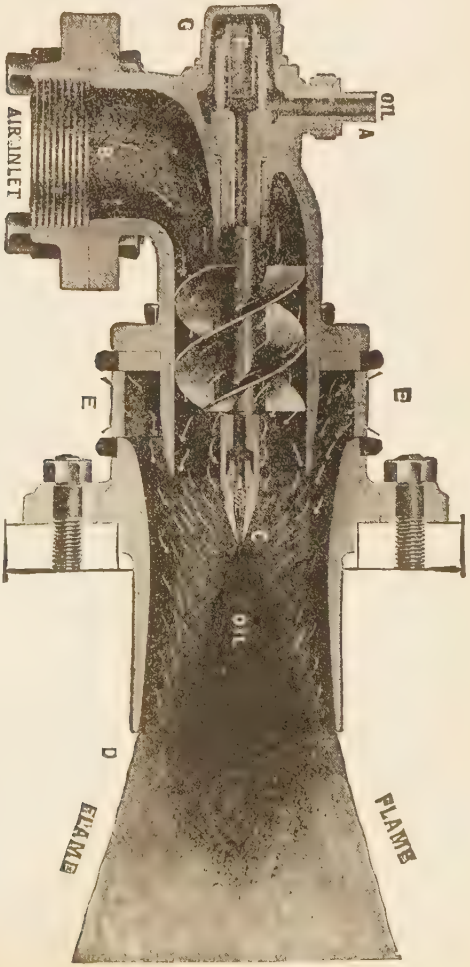


Fig. 35

air as it issues and mixes with the oil, thus forming a spreading cone.

Referring again to Fig. 35, *A* is the attachment for the oil; *B*, air inlet; *C*, discharge nozzle for the oil, containing also a screw blade for the purpose of imparting a gyratory motion to the oil as it issues from the nozzle. *D* is the discharge end, and *E* is a slide which controls an auxiliary inlet through which additional air is induced by the action from the central air and oil discharge.

To provide against a stoppage by dirt, each burner is fitted, in the end of the oil tube, with a thimble strainer *F*. After taking off the cap *G*, the strainer *F* can also be cleaned out, for the whole tube can be readily removed, and also the spray nozzle *C* examined, by unscrewing and taking apart.

The oil supply to the burner must be under pressure of not less than thirty-five pounds, at which pressure a sufficient atomization of the oil is reached.

The regulation of the burner is had through difference in pressure. Thus the pump can be set to discharge under a permanent pressure at any point between thirty-five and 150 pounds, which variation is sufficient for all practical purposes.

This burner is used with an air blast supplied by a rotary pressure blower. Usually an air pressure of less than one pound is sufficient for the atomizing process. The volume of air thus passing into the furnace is intended only as a part of the air required for combustion, the balance being admitted through openings especially provided for the purpose.

Re-heaters. While the heating of the oil in the vicinity of the pump suction in the storage tanks is easily accomplished by means of heating coils installed

therein, the re-heating, or super-heating requires apparatus of a different design.

Various devices are in use for this purpose. A very ingenious and effective heater for fuel oil is the film heater, a sectional elevation of which is shown in Fig. 36. In this heater, which is manufactured by the Schutte & Koerting Co., the oil is forced in a thin film between two steam-heated surfaces of such shape that the oil is continually being mixed or agitated.

Referring to Fig. 36, the oil enters at *EO* and passes up and through the film, leaving the heater at *DO*. Steam enters the heater at *ES*, and by means of the branch pipe is carried between the shell and the outer tube and also inside the inner tube, the condensed water draining off at *DS*, thus giving the desired counter-current which adds materially to the efficiency of the heater.

The inner tube is easily removed for cleaning, or the plugs *F* and *F* may be removed and a steam nozzle inserted at one end and high pressure steam blown through the film space which will in a few minutes clean out all accumulated oil or sediment held in the film. This heater may be placed in any position, either horizontal or vertical as found most convenient.

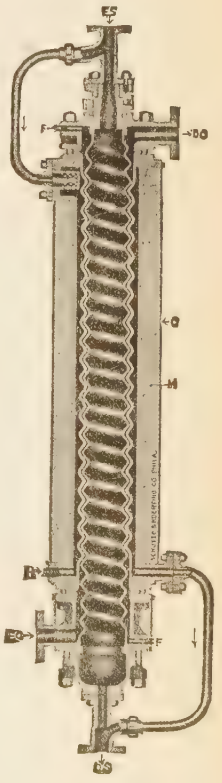


FIG. 36
SCHUTTE & KOERTING FILM OIL HEATER

Oil Pumps. The handling of high-gravity fuel oils, running from 30 Baumé up, and quite liquid in consistency, is usually best accomplished by the use of an ordinary duplex pump fitted with brass-ring packing in the pump pistons and brass valves and special oil-proof gaskets in the pump cylinder joints to overcome the solvent action of the oil. For heavy viscous oil, like Mexican crude, the ordinary duplex piston pump is suitable, provided a size sufficiently large is selected to keep the oil velocities through the ports and passages and valve seats in the pump cylinders down to a minimum.

The type of pump ordinarily used for pumping fuel oil to burners, where the oil pressure will not exceed 150 pounds per square inch, is a pump similar to the Blake-Knowles Duplex piston-pattern pump. In selecting this type for low-gravity oils, it is advisable to use nothing smaller than a 4½x2½x4 inch size, as the ports and valve passages in small pumps are too restricted to operate successfully with oils of this character, and it might be well to remember that the four, five and six-inch stroke duplex fuel-oil pumps have valve areas equivalent to 35 to 40 per cent of the piston area and should operate at speeds not to exceed thirty to forty single strokes per minute each piston, and ten and twelve-inch stroke pumps, twenty-five to thirty single strokes.

Fig. 37 shows a very convenient outfit for handling fuel oil. It consists of two pumps set on a cast iron stand and equipped with oil heater, strainers, by-pass valves, thermometers and gauges.

With this arrangement one pump is operative and the other is a reserve. A pressure governor or regulator is set for the oil pressure and controls the pump

automatically. These systems are usually used in connection with low-pressure burners of the steam or air atomizing type. In rare instances, the single-cylinder

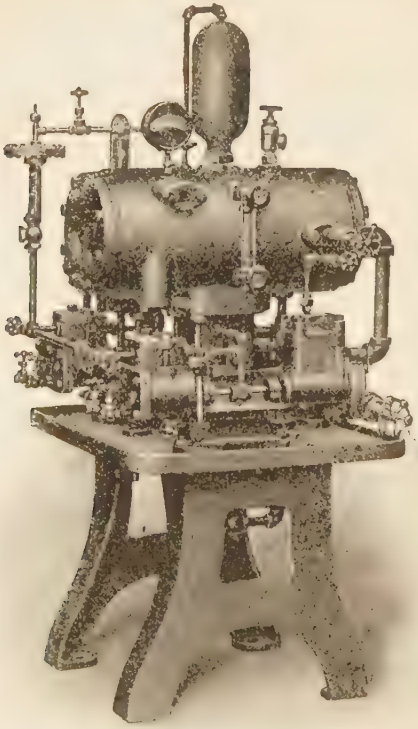


FIG. 37
FUEL OIL PUMPING OUTFIT

pump is used, although in this country the duplex pump is favored.

For high-pressure fuel-oil systems it is necessary to use pumps designed for operating against oil pres-

sures up to 200 pounds per square inch. For horizontal work, the duplex-pattern pump of the valve-plate style makes a very efficient and satisfactory pump for the service. For naval installations or for marine work where floor space is limited, the vertical duplex Admiralty pump attached to the bulkhead is usually employed. This type of pump is good for 300 pounds maximum oil pressure.

The simplex pump, either vertical or horizontal, has better suction qualities than the duplex, and if properly fitted up with air chambers on both the suction and discharge sides will give practically as constant and steady flow of oil as will the duplex type. The increased suction capacity of the simplex pump is due to long stroke and large valve diameter as compared with the duplex type of equal capacity. Large valve diameters are especially favorable to the free flow of heavy or viscous oils. In attaching a suction air chamber to a pump it should be connected so as to allow the direction of flow through the suction to cushion against the air or gas contained in the suction chamber, rather than have the flow pass directly into the suction opening of the pump.

Fig. 38 shows two designs for the attachment of suction air chambers to fuel oil pumps.

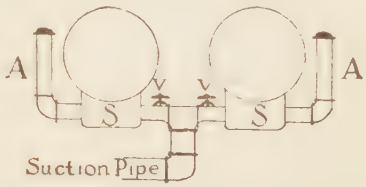


FIG. 38

DESIGN FOR ATTACHMENT OF SUCTION AIR CHAMBER TO FUEL OIL PUMP

Rotating Plunger Pumps. While the reciprocating pump has been largely used for pumping oil to burners, there are several makes of rotating plunger pumps which give satisfactory results for this purpose, and have the advantage of giving a much steadier oil pressure. These pumps utilize the well-known principle of the rotary engine, and experimental tests have shown them well qualified for use with mechanical-atomizing oil-burning installations.

Strainers. The comparatively small orifice of the mechanical sprayer necessitates special care being taken in straining the oil. Most makes of burners are fitted with individual strainers; that used in the Peabody burner being wrapped with three turns of brass netting of forty-to-the-inch mesh, fastened with No. 20 gauge brass wire. This is easily renewable when necessary.

SOOT CLEANERS

The formation of soot cannot be prevented under the conditions ordinarily existing in a boiler setting. It is formed during the process of combustion by contact of the gases with the heating surfaces of the boiler before combustion is completed, and is in fact, uncombined carbon which adheres to the heating surfaces in the form of soot. It is one of the greatest obstacles to the proper transmission of heat. The following table shows in a condensed form the loss in heat conductivity due to soot deposits on the heating surfaces of a boiler:*

*Jacques Abady in a lecture before the Nottingham Guild of Mechanical and Electrical Engineers.

Thickness of soot—	Per cent loss
Clean pipe	0.0
$\frac{1}{32}$ inch	9.5
$\frac{1}{16}$ inch	26.2
$\frac{1}{8}$ inch	45.2
$\frac{3}{16}$ inch	69.0

In order to determine the extent of the losses due to soot deposits, the Engineering Department of the University of Illinois conducted a series of tests with the following results. Three series of tests were made, each covering a period of five days, as follows:

First series—Soot is allowed to remain on tubes.

Second series—Tubes cleaned each morning.

Third series—Soot allowed to remain on tubes.

	First Series	Second Series	Third Series
Equivalent evaporation from and at 212° F. per lb. of dry coal	6.20	7.04	6.23
Dry coal per sq. ft. of grate surface per hour	13.40	9.09	13.40
Temperature of escaping gases	627.00	546.00	698.00

	Per Cent
Increase in evaporation due to clean surfaces	13.0
Decrease in coal burned per sq. ft. of grate per hour due to clean surfaces	32.16
Lowering of stack temperature, due to clean surfaces	17.59

From the preceding data it is plainly evident that the daily cleaning of soot deposits from the boiler heating surfaces is a necessity, if efficiency is to be maintained, and economy in fuel is any object. The problem therefore with which the engineer in charge is confronted is the best method of accomplishing this work. There are various types of soot cleaners or blowers, but in all cases the work is performed by a jet of compressed air or steam discharged through a nozzle of some form and applied in such a manner that it will

FIG. 39.

DESIGN AND LOCATION OF ELEMENTS FOR VULCAN SOOT CLEANER AS APPLIED TO A WATER TUBE BOILER

dislodge the coating of soot from tubes and other heating surfaces. Although compressed air is effective in soot blowing, its use is not advisable, owing to its

chilling effect and the resultant unequal strains thus brought to bear upon heating surfaces. Another disadvantage in connection with compressed air is that it is not at all times available in many boiler rooms, while steam may always be procured while the boilers are in operation.

The hand soot blower consists of a steam tube of proper size to enter the setting of a water tube boiler or the tubes of a return tubular boiler. The tube is fitted with a steam nozzle at one end, and having the other end attached to a flexible steam hose which in turn is connected to the steam main. This type of soot cleaner is still in use in many boiler rooms, but in order to use it the doors must be opened to insert the tube, consequently there is more or less loss due to cold air entering the gas passages, while the steam pressure and draft are also affected. Operating a hand soot blower is also very hard and disagreeable work and requires considerable time.

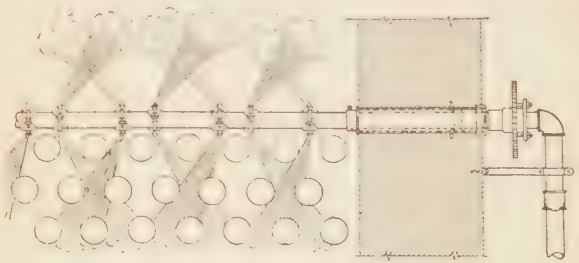


FIG. 40
VULCAN SOOT CLEANER—SINGLE ELEMENT WITH TWO SETS OF NOZZLES

The disadvantages attendant upon the use of the hand soot blower can be practically eliminated by using a mechanical soot cleaner located inside the

boiler setting, thus avoiding all chilling effects, besides shortening the time required for the cleaning process to one-third that required by the hand blower. Another great advantage gained by the use of a good mechanical soot cleaner lies in the fact that the cleaning may be done more frequently, and with practically no interruption to the normal operation of the boiler. The action of the cleaner is so rapid that the soot as it is dislodged, is at once swept from the boiler, and does not again settle on the heating surfaces. There are various makes of mechanical soot cleaners designed so as to be adaptable to any type of boiler; the important object to be sought being that the nozzles shall be so located that the steam or air jets will strike all parts of the heating surfaces where soot is likely to be deposited. Steam is the cleaning agent usually employed, and each blower should be equipped with a drip by means of which the supply line can be made clear of all water before starting to operate the cleaner. In the application of mechanical soot cleaners to water tube boilers, especially those types in which the tubes are either horizontal or inclined, as for instance the Heine, the Babcock & Wilcox, and others, it was early discovered that a cleaner element which discharges steam directly downward onto a bank of staggered tubes, is ineffectual in cleaning the surface because the tubes themselves muffle the penetration of the jets of steam. This led to the invention of the Vulcan principle, which consists in directing the steam jets along the diagonal paths at about 60 degrees to the horizontal, as shown in the drawing Fig. 39. Each element is mounted on bearings and rotated by a chain and sprocket wheel outside the setting (see Fig. 41) so that the jets of steam are caused

to sweep through a wide arc, concentrating their cleaning and scouring action successively on each portion of the tube surface. Two elements are usually located in each pass; one blows down one set of diagonals in

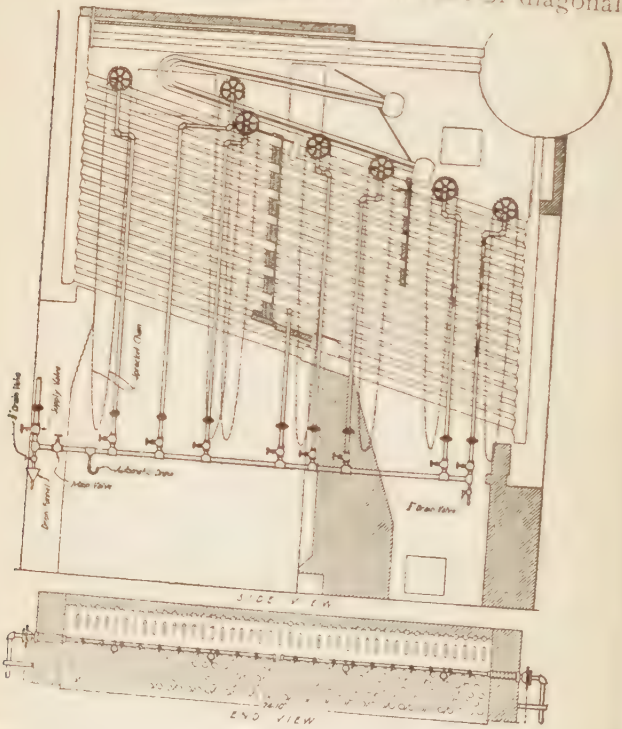


Fig. 41

BABCOCK & WILCOX BOILER AND SUPERHEATER EQUIPPED WITH VULCAN SOOT CLEANER

one direction, the other element down the other set of diagonals in the other direction, as shown at the top and bottom drawings of Fig. 39. The same results may be obtained by locating both sets of nozzles

one element for each pass as shown in Fig. 40. Where conditions are suitable this type is to be recommended.

The operation of the Vulcan soot cleaner is practically the same for all water tube boilers. The valves are blown in rotation and generally in the same direction as the draft. Cleaning every six or eight hours is generally best practice.

The fundamental principle of cleaning this type of boiler consists in discharging jets of steam into the clear diagonal passages between the tubes and then, by slowly rotating the element, causing the steam jets to sweep back and forth, so that all of the power of the steam is concentrated for a short time on every section of the heating surface.

For some installations rotating and stationary elements are combined. The design recommended in any particular case depends on how the boiler is baffled, and also upon local conditions. Fig. 42 shows a Vulcan soot cleaner, as applied to a Stirling boiler with the standard arrangement of baffling, and equipped with a superheater. Each element is numbered, and it will be noticed that element No. 1 is located in the midst of tubes above the mud drum and is protected from the heat by the short baffle placed between the first and second rows of tubes. This element is rotatable, and cleans upward over all of the first pass. The surface is also cleaned downward by element No. 2 located above the circulating tubes, where it is fully protected from the heat, and yet has a wide range over the first and second passes and the superheater. The dotted lines show the operating chains.

The general design and principles of the Vulcan soot cleaner as here outlined are applied to other standard types of boilers, both water tube and fire tube, or return

tubular. For application to the latter it is made in two types: (a) for installation at the rear end, (b) for installation at the front of the boiler.

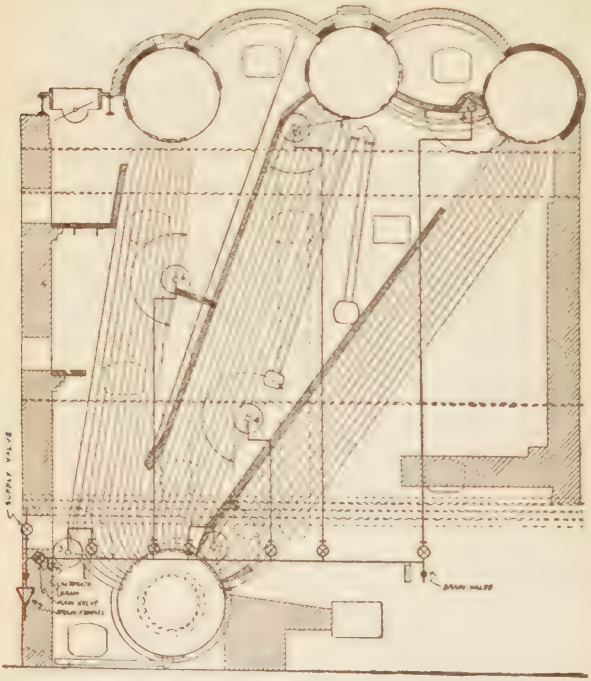


FIG. 42
STIRLING BOILER EQUIPPED WITH VULCAN SOOT CLEANER

Fig. 43 shows the rear end design; it consists of a moveable element which swings in a half circle, across the entire tube area. The element carries nozzles that discharge powerful jets of steam directly into every tube, blowing with the draft. The steam jets are so arranged that it is impossible for a tube to be passed over without cleaning by some one of the jets.

The cleaner is operated by pushing it out from its recess in the wall, where it lies protected from the gas current when not in operation, and rotating it by means

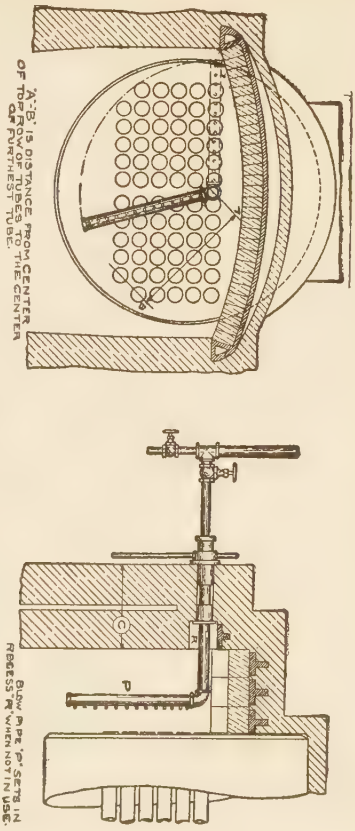

"A-B" IS DISTANCE FROM CENTER OF TOP OF OUTSIDE TUBE TO CENTER OF FURTHEST TUBE.
A
B
C
R
P
BLOW PIPE "P" SETS IN RECESS "R" WHEN NOT IN USE.

FIG. 43 VULCAN SOOT CLEANER APPLIED TO REAR END OF A RETURN TUBULAR BOILER

of a handle from outside the setting and so moving the nozzle arm a few inches at a time, allowing a few seconds at each stop, for the steam to drive all soot from the tubes The whole operation of cleaning the boiler

takes only a few minutes and should be repeated every few hours during the day.

A "front end" Vulcan soot cleaner is shown in Fig 44. This cleaner is very similar to that just described, rotating so as to bring steam nozzles opposite all of the tubes in a boiler and cause a blast of steam to scour out the soot and ash deposits in the tubes. No protecting recess is required as the temperature of the gases is comparatively low at this point, where they are finally leaving the boiler. The cleaner is operated by a chain and sprocket, which is all that appears on the front of the boiler. The apparatus does not interfere with the opening of the smoke doors; these should be kept tightly closed to prevent air leakage.

Two very important features of a mechanical soot cleaner system for water tube boilers which should not be overlooked at the time of installation are (1) protection of the outside valves and piping from internal corrosion; (2) prevention of water getting into the main header or steam supply. Internal corrosion is caused as follows: Upon closing one of the operating valves the steam contained in the riser condenses, forming a partial vacuum, and as the piping is now closed to the atmosphere, the pressure balance can only be restored by the piping being filled with the sulphur-laden furnace gases. The sulphur dioxide in these gases enters into combination with the moisture on the inner walls of the pipe, forming sulphuric acid, the presence of which causes serious erosion of the metal.

This trouble is entirely corrected in the Vulcan system by the use of small automatic ball-check valves on all risers from the main header. When the valve on any riser is closed the ball-check instantly falls away

from its seat, opening a minute orifice, through which sufficient air is admitted to restore the balance of pressure within the riser, and prevent the entrance of furnace gases. In order to guard against the possibility

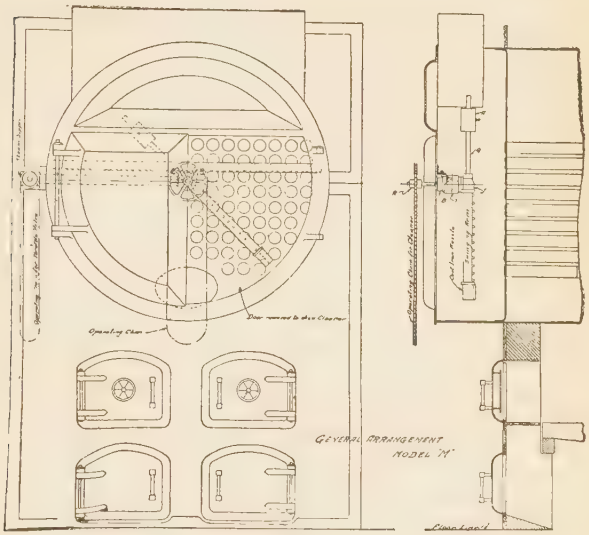


FIG. 44
VULCAN SOOT CLEANER APPLIED TO FRONT END OF A RETURN TUBULAR BOILER

of danger from the accumulation of water in the main header; automatic drain valves are located at the most available points. In addition to these, the system is also well provided with ordinary drain valves as will be seen by reference to Figures 41 and 42.

Other types of mechanical soot cleaners are in use also, as for instance the Diamond, the construction and operation of which resembles to some extent the Vulcan system; the Bayer soot blower, the action of which

for vertically baffled horizontal water tube boilers is based upon the projection of oscillating steam jets into and laterally across the spaces between the rows of tubes, in planes parallel to the pitch of the tubes. The Marion flue blower is provided with a nozzle having three or four openings, according to the size of the boiler, and by means of a valve mechanism, steam is alternately turned from one opening to another. Each opening has a different angular position, and the steam jet is directed on one-third or one-fourth of the flues at a time. Steam is diverted from one opening to another by raising the spring connected to the valve stem and moving it to the next notch on the hand-wheel. Set even with the inside brick wall is a large cup-shaped casting which protects the nozzle from the heat of the combustion chamber.

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